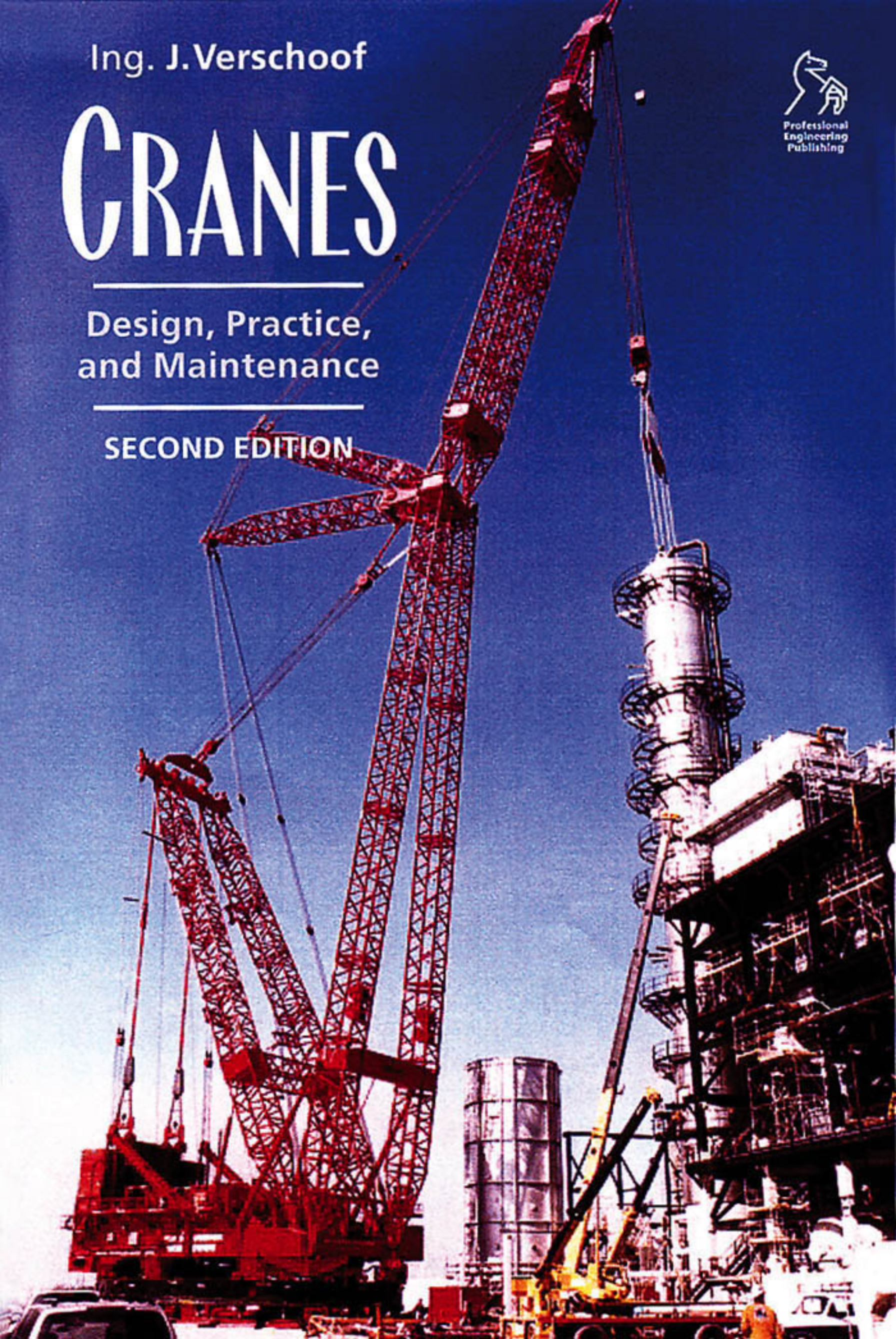
Ing. J. Verschoof

CRANES

Design, Practice, and Maintenance

SECOND EDITION

Professional Engineering Publishing



Cranes – Design, Practice, and Maintenance

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Cranes – Design, Practice, and Maintenance

(2nd Edition)

by

Ing. J. Verschoof

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Contents

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It is most important to do the things right;
above all it is most important to do the right things

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Foreword

This book looks at all types of cranes and deals with container and grab unloader cranes in some detail.

The objective has been to give a general overview of this very wide subject and then to provide positive, practical guidance to anyone involved in the design, specification, selection, or operation and maintenance of cranes.

A crane is often a very large and complex piece of equipment and this book analyses many of the individual components and design features that can be found on a typical crane. Components such as brakes, wire ropes, electrical drive systems, automatic sensors, wheels, rails, buffers, cable reels, festoons, hoppers, overload preventers, and anti-collision systems are discussed in some detail along with advantages and disadvantages of various component types. This book also shows how to correctly 'size' and calculate a number of these components.

Furthermore, various design features and preferred solutions are discussed such as the effect of wind on cranes, design standards, welding methods, structural design and fatigue calculations and, finally, maintenance.

Anyone involved in crane specification, selection, operation, and maintenance should find the level of detail invaluable when considering potential problem areas, especially some of the 'rule of thumb' recommendations. Crane manufacturers facing problems of detail design will find this book useful in understanding the overall background and environment in which cranes work.

C J Hall BTech, CEng, MIMechE
Chief Engineer
Thamesport, UK

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Foreword to the Second Edition

I am pleased to present the second edition of *Cranes – Design*, *Practice*, *and Maintenance*. I have been very heartened by the response to the first edition of the book and I hope that those who use this updated version, find it even more helpful in their work.

In this new edition, I have brought the section on legislation up-to-date, included a number of exciting new technical developments, and described new equipment that provides advances to the use of cranes. Some important material on new systems for materials handling is added. Many new photographs have also been included.

As this book goes to press, I am working with the publishers to make further information available electronically. Further information can be found via the publisher's website at www.pepublishing.com. We hope to show new and fascinating designs of some crane builders, designers, consultants, and information from academic sources.

I hope you enjoy using this book.

J Verschoof
11th July 2002

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Preface

It is the intention of this book to give all those who have anything to do with hoisting cranes and hoisting equipment a clear source of information on the most important and interesting aspects of this vital and complex equipment.

Who will find this information useful?

This book is designed to help all those involved in the use, maintenance, purchase, specification, design, and construction of cranes. These complicated and expensive devices deserve specialist attention and require a detailed understanding of their workings. Directors, consultants, technicians, engineers, project managers, maintenance contractors, and those involved in the design of cranes will find useful material here.

The author aims to provide understanding of the construction, components, and calculations required for the safe and efficient operation and designs of cranes.

The analyses, formulae and calculations included here provide the first principles, theory, and numerous examples, so that the reader, having understood the basic principles and the methods of calculation, can go on to calculate or recalculate certain problems. It is hoped that the years of experience in the crane industry, which have led to this publication, will assist those grappling with practical problems today.

Numerous photographs and diagrams are included, showing various pieces and types of equipment in action. Above all, this volume is for the practising engineer whether working with this equipment every day, or occasionally coming into contact with cranes.

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Chapter 1

Introduction

1.1 History

The problem ‘how to lift a load’ is as old as humankind. From the earliest times people have faced this problem. Dragging and carrying were employed until the invention of the wheel, when carts, which could be driven or pulled, were built. People worked together to lift loads or to move heavy objects.

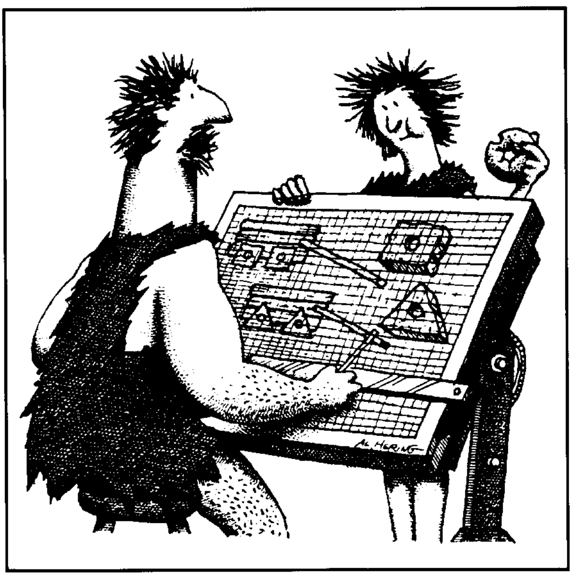


Fig. 1.1.1 Creative problem solving

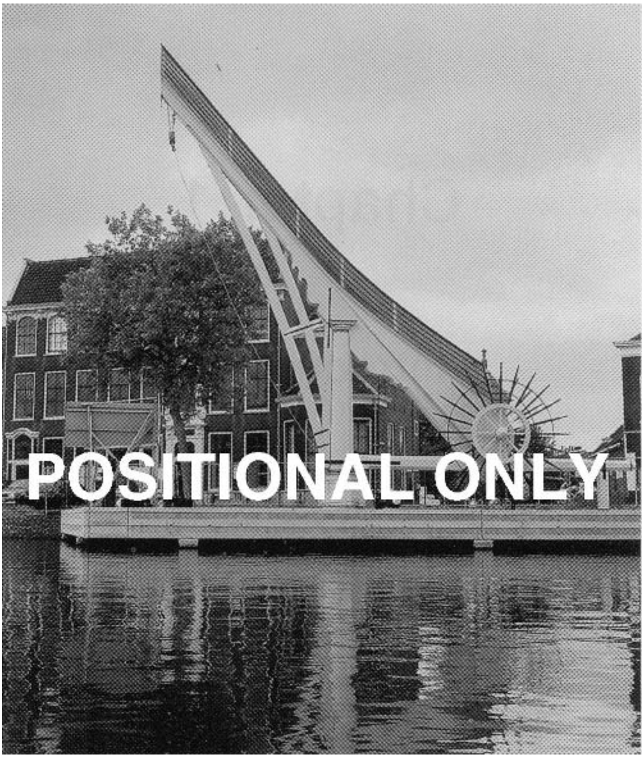


Fig. 1.1.2 Wooden crane

As civilization developed and labour could be organized, structures such as the pyramids were built. The Egyptians lifted and moved enormously heavy stone blocks in the construction of these vast tombs. Theories vary, but it is believed that they used a sort of cradle and employed the forces of momentum and equilibrium to manoeuvre the otherwise impossibly unwieldy blocks. Horses and other animals were harnessed to provide the power to deliver the motive force for lifting and moving heavy objects.

The Middle Ages saw the development of wooden-built slewing cranes that are well known from the harbours of the day. Breughel, the painter, depicted huge Belgian horses in some of his paintings, drawing loaded sledges and powering wooden cranes to hoist large loads. Some hoisting mechanisms were driven by a number of men walking a treadmill or by capstans. Huge wooden cranes were constructed for hoisting masts and other weighty items in shipyards.

Fig. 1.1.3 Driving the hoist of Fig. 1.1.2

Man gradually developed the technologies of using water, steam, and other power sources. For example, James Watt introduced the efficient use of the steam engine. As greater and greater forces were generated it became necessary to give a dimension to this power so that it could be measured and described. The term ‘horsepower’ was coined and this represents 75 kgm/sec.

Electricity followed as another manifestation of power. The development of engineering science to generate, transmit, and store electrical power gave rise to great advances in the complex application of man-made power to move objects. The development of manufacturing techniques in the making of steel-plate and profiles, the knowledge of how to bolt and rivet, and other systems to construct large and strong structures gave rise to the possibility of the manufacture of water-driven, steam-driven, and electrically driven cranes.

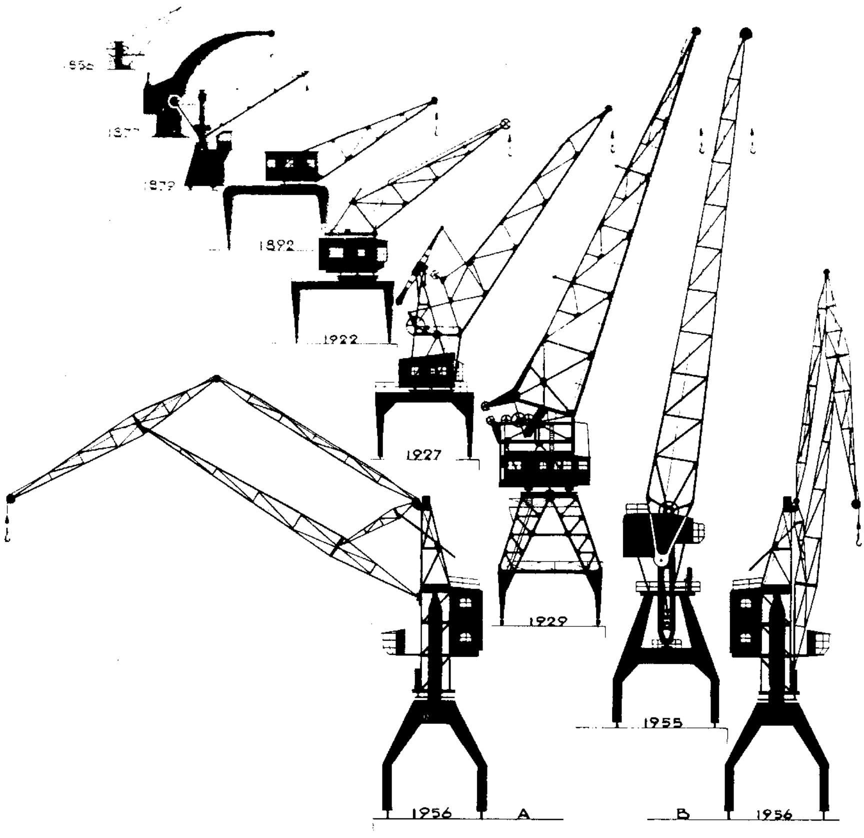


Fig. 1.1.4 The development of slewing level luffing cranes from 1856–1956

Enormous advances now mean that huge loads can be lifted by offshore- and derricking- and slewing cranes where hoisting capacities of 2000 tons or more are routine. Figures 1.1.4 and 1.1.5 illustrate the development of cranes over relatively short periods of time and show the vast differences in size and lifting capacity. Figures 1.1.6 and 1.1.7 show typical cranes that are in use today.

Acknowledging the great strides made by our predecessors, in advancing the technology, science, and engineering which has transformed our ability to lift, hoist and move vast objects of huge mass, and looking forward to the challenges and problems of today and tomorrow, the following saying is appropriate:

Hats off for the past, coats out for the future.

(Winston Churchill)

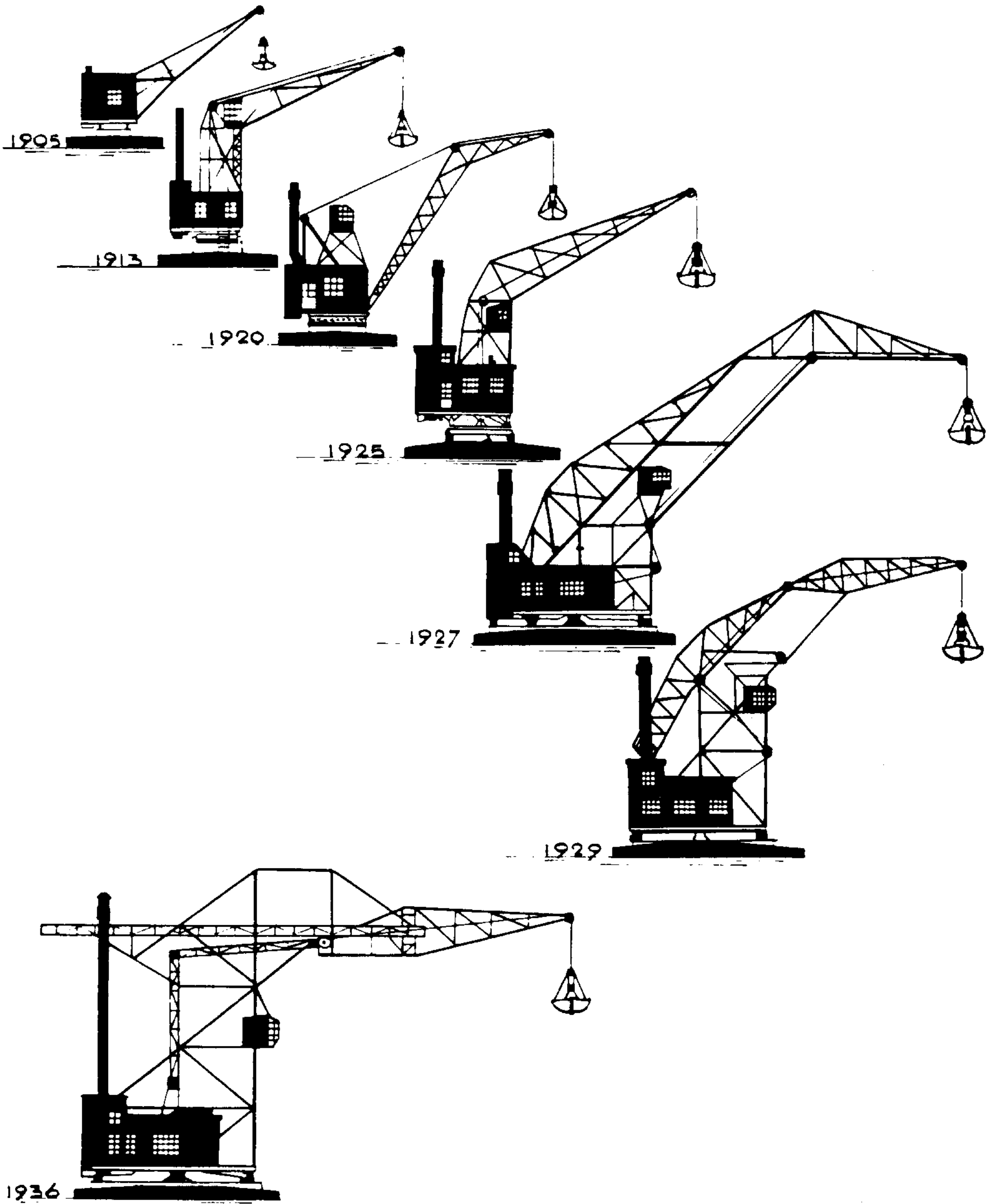


Fig. 1.1.5 The development of floating cranes 1905–1936

Fig. 1.1.6 100 t mobile 'all-purpose' crane

1.2 Power

Archimedes said:

Give me a lever long enough and a place to stand and I will lift the world.

While, in theory, this would seem to be true, in practice it is not of course possible. There are real practical limitations. How much work can a man do?

Fig. 1.1.7 Lemniscate grabbing crane

– Using a handle: maximum 8–12 kilogram force on a stroke of approximately 0,4 m

  So $A = K \times S = 8 \times 40 = 320$ kg cm = 32 Nm

– Using a hand driven winch: during a maximum of approximately 15 minutes maximum 8–12 kilograms on a radius of approximately 0,4 m and with a circumference speed of approximately 1 m/sec

– The pulling force on a rope for hoisting by one man is about 25 kg during short periods.

Power consumption by one man is approximately:

– During normal walking	30 Watt
– During fast walking	60 Watt
– During fast running	160 Watt
– During working in the garden	140 Watt

(1 Joule = 1 Wattsec)

The power driving a fast modern hoisting winch can reach up to 1700 kilowatts or more. Handling, driving and steering a powerful modern crane is a complicated and technically demanding operation. In later sections we examine some of the issues and problems involved.

Note

Please note that in all the calculations offered in this book the following standard European notation is used, instead of the English/American notation.

Example

$\eta = 0{,}90$ instead of $\eta = 0.90$

$$M = \frac{46 \cdot 81{,}95}{2} \text{ instead of } M = \frac{46 \times 81.95}{2}$$

$$N_3 = \frac{12{,}74 \cdot 2}{0{,}9} \text{ instead of } N_3 = \frac{12.74 \times 2}{0.9} \text{ or } N_3 = \frac{12.74 \cdot 2}{0.9}$$

This is done in order to make the formulae somewhat clearer in print and for ease in reading.

Fig. 1.2.1 2500 t offshore crane

All dimensions are given in metres or millimetres. When tons are mentioned, these are metric tons (1t = 10 kN = 1000 kg.) Gravity is normally calculated as $g = 10\,\mathrm{m/sec^2}$ instead of $g = 9{,}81\,\mathrm{m/sec^2}$.

A number of manufacturers are exampled and mentioned in this volume, this does not mean that these are particularly recommended. Nor does it mean that other equally respectable and technically sound manufacturers and equipment are not recommended. The examples are merely a matter of the author's experience and personal familiarity. All those concerned with this type of equipment should make their own decisions as to which equipment is most suitable to fit their purpose.

1.3 Some types of cranes and lifting equipment

There are many types of cranes, some of which are illustrated in Figs 1.3.1 to 1.3.13.

There is now a choice of drives, electric or hydraulic. The main electricity supply may be directly from a power station. This power is usually delivered as a medium or high-tension voltage, typically 10 kV, which will be transformed into, for example, a 500 Volt supply, on the crane itself using the crane's own transformer.

Fig. 1.3.1 Container quay cranes

Fig. 1.3.2 Automatic stacking cranes

Diesel engines are also used as the motive power for cranes and transporters. These engines power the generators and oil pumps. The generators drive the electric motors and the oil pumps serve the oil-driven motors or hydraulic cylinders for the different mechanisms.

Straddle carriers, which lift containers and transport them at high speeds, are grouped with lifting equipment, along with container lifters which are mounted on trucks and barges. Reach stackers, draglines, and many others also belong to this family.

Fig. 1.3.3 Bulk unloader

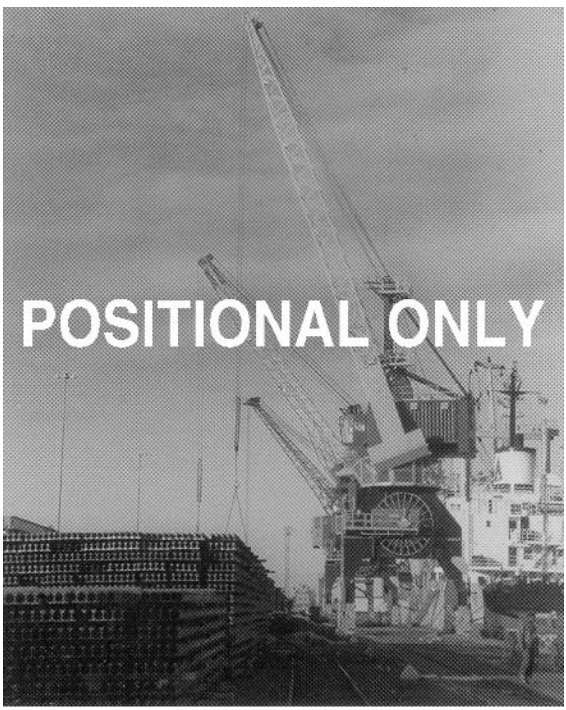


Fig. 1.3.4 Level luffing cranes

Fig. 1.3.5 Bock crane

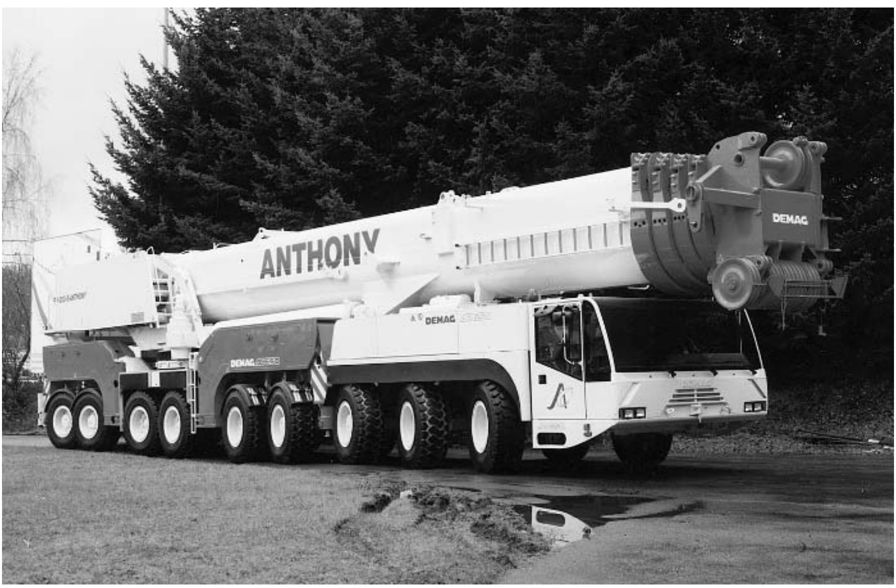


Fig. 1.3.6 Autocrane for heavy loads

Fig. 1.3.7 Ringer crane

Fig. 1.3.8 Floating derrick

Fig. 1.3.9 Two offshore cranes; each for 4000 t (By courtesy of Heerema/IHC Gusto Eng. BV)

Fig. 1.3.10 Tower crane

Fig. 1.3.11 Container mover

Fig. 1.3.12 Excavator

Fig. 1.3.13 Shuttle carrier

1.4 Capacities, number of cycles, cycle-time

Container quay cranes

In the container business, the containers are referred to as TEUs. A TEU is a 20 ft equivalent unit. A 20 ft container is one TEU, and a 40 ft container is two TEUs. In converting the number of TEUs to the number of ‘moves’ it can be assumed that a ratio of 1 : 1 of 20 ft to 40 ft containers does not exist today. Therefore, a TEU factor of 1,5 is produced. As the proportion of 40 ft containers seems to be increasing, the TEU factor will rise, and in the near future it will be reasonable to assume a TEU factor of 1,6.

In container handling operations managers often say that they expect and achieve a high number of 'cycles' or 'moves' per hour. Theoretically the time a duty cycle takes can be calculated, but factors that can disturb or affect efficiency must also be taken into account. Many operators state that they would like to calculate using a capacity of 100–125 containers per hour per ship using a maximum of three to four container quay cranes, working together loading or unloading one ship.

Example

– Container vessel	4000 TEU
– Number of containers with a TEU factor of 1,5	$4000 \div 1{,}5 = 2666$ cont.
– Number of containers to be *unloaded* in the particular harbour – 60%	$2666 \cdot 0{,}6 = 1600$ cont.
– Assumed number of containers which have to be *loaded*	= 1200 cont.
– Total number of containers which have to be handled	= 2800 cont.
– Total of the time which the vessel is to be allowed to stay moored	24 hours
– Needed as average hour capacity	$2800 \div 24 = 117$ cont./hr

What are the disturbances and how great is their impact?

The following disturbances must be considered.

Average operation time over a number of vessels:
(Normal, real operation time, without disturbances = 100%)

– Time for lashing/unlashing	–
– Time to unlock/lock semi-automatic container cones	–
– Dealing with hatch covers	–
– Hoisting/lowering the boom	–
– Breakdown of the crane	–
– Break for meals/refreshments	–

– Shift changes	–
– Waiting for transportation ashore	–
– Loss of time due to jammed twistlocks	–
– Delays due to the vessel	–
– Waiting time to start work	–
– Time to examine control seals, any damage, and the CSC plate	–
Total	%

It is vital to be aware that, under certain circumstances, the total of these disruptions can be up to 30–40 percent of the potential operation time. It is often assumed that capacity increases when the movements are automated or semi-automated, but the level of improvement in capacity varies from harbour to harbour and from operative to operative. The capacity of a container quay crane will be greatest when a skilled crane driver is being used. However, people do tire but automation never becomes fatigued in the same way. Therein lies the difference!

In the USA the following productivity measures have been developed by, among others, the National Ports and Waterways Institute. The data here are by kind permission of Dr A. Ashar.

$$\text{Port time} = \text{Port access time} + \text{Terminal preparation time} + \text{Terminal handling time}$$

$$\text{Terminal handling time} = \text{Container moves}/\text{net berth productivity}$$

$$\text{Net berth productivity} = \text{Net gang productivity} \times \text{Average number of gangs}$$

Container moves

While serving a ship a gang may perform a series of direct and indirect activities. The activities are usually qualified by 'moves', the four most common types of which are:

(a) *Load/unload* – the transfer of domestic (import and export) and transhipment boxes between ship and yard;

(b) *Re-handle* – the transfer of transhipment boxes between ship and dock for a later transfer from the dock to the same ship;

(c) *Shifting on-board* – the transfer of boxes between bays (cells) without staging them on dock;

(d) *Hatch opening/closing* – the transfer of hatchcovers between the ship and the dock.

Definitions of times, activities, and quantities

Ship and gang times

The services that a ship receives at a port begin when the ship arrives at the entry buoy and ends when the ship passes the buoy on its way out, after finishing loading/unloading its cargo. The actual handling of cargo is performed by one or more gangs, each using a shore-based or ship-based crane. The times and the activities are generally divided into those related to the ship itself, and those related to the gangs or cranes working the ship. The ship handling process involves many activities and times. For simplification, the times are incorporated into six functional categories; three related to ships and three to gangs.

Ship times include:

(a) *Port time* – the buoy-to-buoy time; the total time that the ship spends at a port, including waiting for a berth, documents, pilot, tugs, delays due to bad weather, etc.

(b) *Gross berth time* – the first-to-last line time, the total time that a ship is at berth, including ship preparations, waiting for documents, gangs, beginning of shift, change of shifts, availability of cargo, etc. and the major delays during work due to equipment breakdowns, bad weather, etc.

(c) *Net berth time* – the first unlash-to-last lash-time, or the working time of a ship at berth, during which gangs load/unload the containers and perform related activities such as lashing/unlashing, placing/removing cones, opening/closing hatchcovers, etc. The net berth time includes minor during-work interruptions due to unavailability of cargo, equipment breakdowns, etc.

Gang (crane) times include:

(a) *Gross gang time* – the time that a gang is available (assigned) to work a ship and for which the gang is paid, including waiting times before and after work (stand-by) and interruptions during work.

(b) *Net gang time* – the time that a gang is actually working, including handling boxes and performing other, indirect activities, along with during-work minor interruptions.

(c) *Net/net gang time* – the same as net gang time, but only including the time spent handling containers.

Ship and gang productivity

Ship productivity includes three measures:

(a) *Port accessibility* – the difference between port time and gross berth time. This measure reflects:

- the geographical situation of a port, mainly the distance and navigation conditions on the access channel;
- availability of pilots and tugs;
- availability of governmental agencies responsible for clearing ships, crews, and cargo; and
- availability of berthage.

(b) *Gross berth productivity* – 'moves' (boxes) transferred between the ship and the dock/yard, divided by ship's gross berth time – the difference between the first and the last line. This measure reflects the shift structure and labour situation.

(c) *Net berth productivity* – the same as gross berth productivity, but using net berth time. This measure reflects the number of gangs (cranes) assigned to the ship and the net gang productivity (see below).

Gang productivity also includes three measures:

(a) *Gross gang productivity* – 'moves' divided by gross gang time. This measure reflects labour contract, especially regarding idle 'stand-by' times at the beginning, during, and end of shifts 'early finish'.

(b) *Net gang productivity* – the same as gross gang productivity, but using net gang time. This measure reflects necessary, although non-productive, that is not producing 'moves', activities such as handling hatch covers, shifting boxes, on-board (cell-to-cell) 'moves', inserting/removing cones, etc.

(c) *Net/net gang productivity* – the same as above but using the net/net gang time. This measure, also called 'pick rate', reflects the technical capability of facilities and equipment, along with the proficiency of the labour in operating them and the competence of terminal management in planning and controlling them.

Since all times are usually measured in hours, the productivity measures are all expressed in moves/hours.

Grab unloaders

The definition of a grab unloader is a ship-to-shore unloader with a built in hopper. The maximum capacity of these unloaders can be from 1500 tons per hour up to 6000 tons per hour. Unlike a continuously

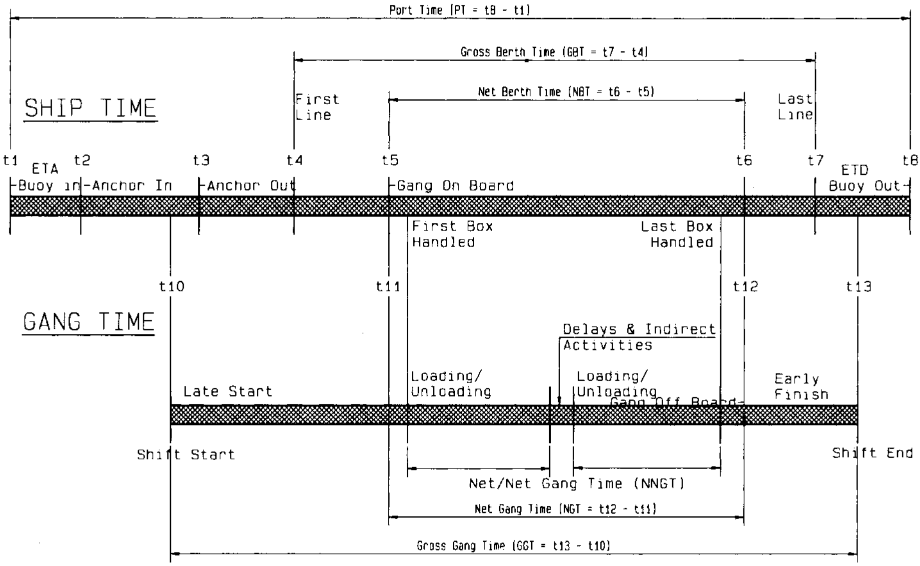


Fig. 1.4.1 Break-down of ship and gang times (by kind permission of Dr A. Ashar)

running conveyor whose capacity can be easily defined, the definition of the unloading capacity of the intermittently working grab-unloader is less simple.

Different terms are used:

(a) maximum capacity;
(b) free digging capacity; and
(c) average capacity.

Maximum capacity

This is the maximum capacity that can be reached. It depends upon the shortest cycle time, the maximum load of the grab, the skill of the operator, and the shape of the hatch of the ship which is to be unloaded. Operator skill and hatch configuration, are factors which equally affect the free digging and average capacity. In fact a crane driver can maintain this capacity for only a short period of time. The rating of the hoist motors and trolley travelling motors must be designed so that working at maximum capacity does not lead to overloading or overheating that would lead to further loss of potential maximum capacity.

Free-digging capacity

This is the capacity that can be maintained during a certain time, under certain conditions, with a skilled crane driver and takes into account

Table 1.4.1 Definition of productivity measures

Parameter	*Notation*	*Unit*	*Description*
Ship times			
Port time	Tp	Hour	Buoy-to-buoy, including wait at anchorage (for pilot, tug, berth, clearance, weather, etc.)
Gross berth time	Tbg	Hour	First-to-last line, including waiting before/after work (for gang, clearance, etc.)
Net berth time	Tbn	Hour	First-to-last box, when gangs are assigned including minor waiting during work (for stand-bys, meals, breakdowns, etc.)
Gang times			
Gross gang time	Tgg	Hour	Assigned (paid) gang time, including stand-by (for vessel, cargo, equipment, etc.) but excluding meal breaks
Net gang time	Tgn	Hour	Working gang time (first-to-last box), including handling hatchcovers and minor waiting during work (for cargo, equipment, documents, etc., but excluding meal break)
Net/net gang time	Tgnn	Hour	Working gang time handling boxes only
Gang activities			
Ship-to-yard	Sy	Box	Transferring boxes between ship and container yard
Ship-to-dock	Sd	Box	Transferring boxes between ship and dock (re-handle, one way)
Ship-to-ship	Ss	Box	Transferring boxes between cells (shifting on-board)
Ship-to-dock	Hc	Hatchcover	Transferring hatchcovers between ship and dock
'Moves'	Mv	Box	Sy + Sd
Productivities measures			
Port accessibility	Ba	Hours	Tp − Tbg
Gross berth productivity	Pbg	Moves/Hour	Mv/Tbg
Net berth productivity	Pbn	Moves/Hour	Mv/Tbn
Gross gang productivity	Pgg	Moves/Hour	Mv/Tgg
Net gang productivity	Pgn	Moves/Hour	Mv/Tgn
Net–net gang productivity	Pgnn	Moves/Hour	Mv/Tgnn ('pick rate')

the assumed discharge trajectory. It does not take into account any time for shifting the unloader from hatch to hatch, or time for a break, etc. In addition, the type and conditions of the material which has to be transported must be defined. Commonly the starting point of the trajectory of the grab is taken in the middle of the hatch of the ship (x co-ordinate) and the mean low water line (MLW) as y co-ordinate (see Fig. 1.4.2). The end point of the trajectory should be almost at the centre of the hopper. Care should be taken with the hatch opening.

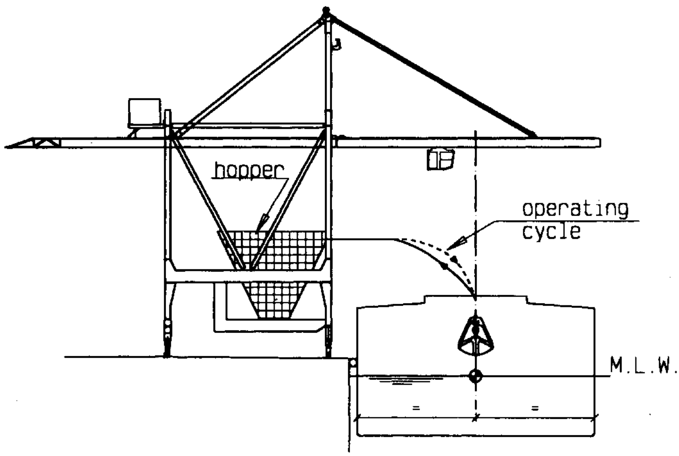


Fig. 1.4.2 Grab unloader

Average capacity

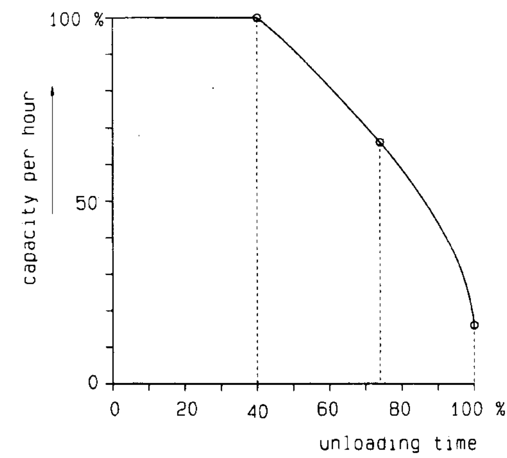
The material-handling manager is also very interested in the average capacity of the unloader. Defining the average capacity per hour is more complicated because it depends upon how the start and finish of the job are measured. It is defined as *The total amount of material that has been discharged during a longer period of time divided by the number of hours*. During this period a great deal of time is lost in shifting the unloader from hatch to hatch, removing and replacing the hatches, meal and refreshment breaks for the working crew, and cleaning up the hatches with a payloader, etc.

Sometimes it is necessary to consider the 'turn-around time' for determining the average capacity. The turn-around time is then considered to be from the moment of mooring the ship or opening the hatches, up to closing the empty hatches or de-mooring the ship.

The average capacity can be roughly indicated as a certain percentage of the maximum capacity or free-digging capacity. The amount of this percentage depends wholly upon the local circumstances, the skill and enthusiasm of the crane driver and the dock personnel, the type of ship, the dimensions of the hatches, the ability to clean up those hatches, and a myriad of other factors.

Larger unloaders are normally semi-automated. The crane driver, sitting in a movable, but stationary cabin, digs in the grab with the help of the controllers. After having hoisted the grab up to a certain position, a knob is pushed which starts the automation. The grab automatically runs towards the hopper, opens, discharges the material, and automatically returns to the point where the crane driver started the automation. The crane driver then takes command again and lowers the grab further to fill it. Duty cycles of approximately 45 seconds can be achieved. However, the average capacity can be as low as 80 or even 60 percent of the free-digging capacity.

For example, the unloading of medium-sized bulk carriers in a particular blast furnace plant with ore, gave, under very good conditions, Fig. 1.4.3.



	Free digging stage	Intermidiate stage	Trimming stage
percentage of the load	50 %	35 %	15 %
percentage of the unloading time	40 %	33 %	27 %
average cap. per hour abt.	1250 t	1060 t	550 t

Fig. 1.4.3 Production scheme

Fig. 1.4.4 Double grab unloader

1.5 The influence of wind and storms

Wind and storms can influence the entire operation of cranes and can even destroy whole cranes. It is vital to make an accurate calculation of the wind forces which the cranes will meet 'in operation' as well as 'out of operation'. In the 1940s the famous Swedish singer Zarah Leander sang a beautiful song called 'Der Wind hat mir ein Lied erzählt' (The wind has told me a song). How true that is.

Wind can be pleasant, strong, a storm, a gale, or a typhoon. The crane must be able to drive against the windforce, which we call the 'operating limit'. This operating limit should be indicated by the company which asks for a tender. In many parts of the world, this will be a $v = 20\,\mathrm{m/sec}$ windspeed (force 8 on the Beaufort scale) which corresponds with a dynamic pressure of the wind of $q = 250\,\mathrm{N/m^2}$. During a storm, this can become $q = 400\,\mathrm{N/m^2}$ or $v = 25{,}3\,\mathrm{m/sec}$ (force 10 on the Beaufort scale means $v = 24{,}5$–$28{,}4\,\mathrm{m/sec}$).

The relation between the wind-speed and the dynamic pressure of the wind is as follows:

$$q = 1/16 \cdot v^2$$

where q = dynamic pressure in kg/m^2 and v = windspeed in m/sec. In the following pages the Rules for the Design of Hoisting Appliances of the FEM 1.001; 3rd edition 1987, 10.01; booklet 2 are quoted. (All extracts of the FEM standards are given by courtesy of the Comité National Français de la FEM in Paris.)

Table T.2.2.4.1.2.1 In-service design wind pressure

Type of appliance	Wind pressure in service (N/m^2)	Wind speed in service (m/s)
Lifting appliance easily protected against wind action or designed for use exclusively in light wind. Erection operations.	125	14
All normal types of crane installed in the open	250	20
Appliances which must continue to work in high winds*	500	28

* For example appliances of type 12a in Table T.2.1.2.5.

Action of wind on the load

The action of the wind on the hook load for a crane which handles miscellaneous loads shall be determined from the relationship:

$$F = 2.5A \times q$$

where

F is the force exerted by the wind on the hook load in N,
q is the in-service wind pressure from Table 2.2.4.1.2.1 in N/m^2
A is the maximum area of the solid parts of the hook load in m^2[(1)]. Where this area is not known, a minimum value of 0.5 m^2 per tonne of safe working load shall be used.

Where a crane is designed to handle loads of a specific size and shape only, the wind loading shall be calculated for the appropriate dimensions and configurations.

2.2.4.1.2.2. Wind out of service

This is a maximum (storm) wind for which the lifting machine is designed to remain *stable* in out of service conditions, as indicated by the manufacturer. The speed varies with the height of the apparatus above the surrounding ground level, the geographical location and the degree of exposure to the prevailing winds.

(1) Where, exceptionally, a crane is required to handle loads of large surface area, it is admissible for the manufacturer to determine a wind speed less than that specified in Table T.2.2.4.1.2.1 above which such loads shall not be handled.

Table 1.5.1 Wind scale and appertaining wind pressure *(q)*

	Wind speed averaged over 10 min. At 10 m height above flat ground or above sea		*Name*		*Wind pressure (N/m²)*
Wind force[1]	m/s[1]	*km/h*[2]	*Above flat ground*	*Above sea*	
0	0,0–0,2	0,0–0,7	Calm	Calm	0–0,03
1	0,3–1,5	1,1–5,4	Light wind	Flat calm	0,06–1,4
2	1,6–3,3	5,8–11,9	Light wind	Flat/freshening	1,6–6,8
3	3,4–5,4	12,2–19,4	Moderate wind	Slight fresh	7,2–18,2
4	5,5–7,9	19,8–28,4	Moderate wind	Moder. fresh	18,8–39
5	8,0–10,7	28,8–38,5	Fairly strong wind	Fresh breeze	40–71
6	10,8–13,8	38,9–49,6	Strong wind	Stiff breeze	73–118
7	13,9–17,1	50–61,5	Hard wind	Hard wind	121–182
8	17,2–20,7	61,9–74,5	Stormy wind	Stormy wind	184–267
9	20,8–24,4	74,9–87,8	Storm	Storm	270–371
10	24,5–28,4	88,2–102,2	Heavy storm	Heavy storm	375–502
11	28,5–32,6	102,6–117,3	Very heavy storm	Very heavy storm	509–662
12	> 32,6	> 117,3	Hurricane	Hurricane	> 662

Notes:

(1) The figure for the wind force is borrowed from the International Beaufort scale, which is originally defined above sea. Depending on the dimensions of waves and the presence of foam crests and such like, the wind velocity can be *estimated* with the help of this scale. Above land therefore it is more difficult, and inaccurate, to work with this scale. However, the indication of the wind force and the relation with the wind velocity, now measured in m/s, are applied internationally both above land, and above sea, for the classification of wind.

(2) The values for the wind velocity in km/h are deduced from those in m/s. There are several methods to calculate the force of the wind on a crane, but it is always necessary to calculate carefully the projected areas; the areas exposed to a direct hit and also all the shielded areas of all parts of the crane which are affected by the wind.

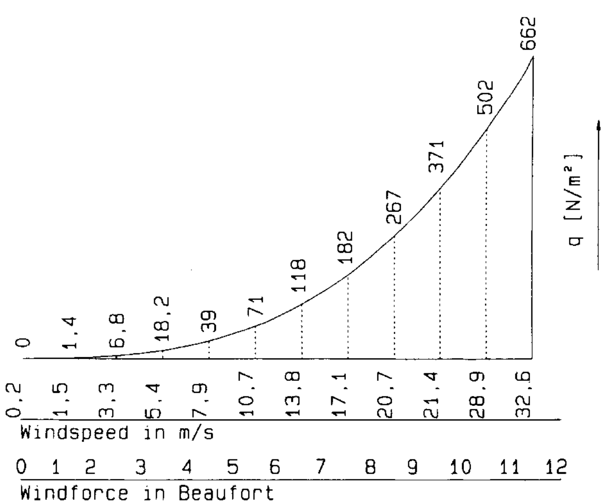


Fig. 1.5.1 The Beaufort scale

For lifting appliances used in the open air, the normal theoretical wind pressure and the corresponding speed, for ''out of service'' conditions are indicated in the Table T.2.2.4.1.2.2.

Table T.2.2.4.1.2.2 Out of service wind

Height above ground level (m)	Out of service design wind pressure (N/m^2)	Approximate equivalent out of service design wind speed (m/s)
0 to 20	800	36
20 to 100	1100	42
More than 100	1300	46

When calculating wind loads for out of service conditions the wind pressure may be taken as constant over the vertical height intervals in Table T.2.2.4.1.2.2. Alternatively, the design wind pressure at the top of the crane may be assumed constant over its entire height.

Where cranes are to be permanently installed or used for extended periods in areas where wind conditions are exceptionally severe, the above figures may be modified by agreement between the manufacturer and purchaser in the light of local meteorological data.

For certain types of appliance of which the jib can be quickly lowered, (such as a tower crane which can be easily lowered by a built-in mechanism) the out of service wind need not be taken into consideration provided the machine is intended for lowering after each working day.

2.2.4.1.3 WIND LOAD CALCULATIONS

For most complete and part structures, and individual members used in crane structures the wind load is calculated from:

$$F = A \cdot q \cdot C_f$$

where

- F is the wind load in N,
- A is the effective frontal area of the part under consideration in m^2,
- q is the wind pressure corresponding to the appropriate design condition in N/m^2,
- C_f is the shape coefficient in the direction of the wind for the part under consideration.

The total wind load on the structure is taken as the sum of the loads on its component parts.

In determining strength and stability requirements of the appliance the total wind load shall be considered.

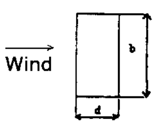
The magnitude of the wind load to be allowed for in the design of a mechanism, in determining the motor and brake requirements for the mechanism and to provide for the safety of the appliance in the wind, are given in the chapter dealing with the design of mechanisms.

2.2.4.1.4 SHAPE COEFFICIENTS

2.2.4.1.4.1 Individual members, frames, etc.

Shape coefficients for individual members, single lattice frames and machinery houses are given in Table T.2.2.4.1.4.1. The values for individual members vary according to the aerodynamic slenderness and, in the case of large box sections, with the section ratio. Aerodynamic slenderness and section ratio are defined in Fig. 2.2.4.1.4.1.

Table T.2.2.4.1.4.1 Force coefficients

			Aerodynamic Slenderness 1/b or 1/D[1]						
Type	Description		≤5	10	20	30	40	50	>50
Individual members	Rolled sections [ ]		1,15	1,15	1,3	1,4	1,45	1,5	1,6
	Rectangular hollow sections up to 356 mm square		1,4	1,45	1,5	1,55	1,55	1,55	1,6
	and 254 × 457 mm rectangular		1,05	1,05	1,2	1,3	1,4	1,5	1,6
	Other sections		1,30	1,35	1,60	1,65	1,70	1,80	1,80
	Circular sections where:								
	$D \cdot V_S < 6\ m^2/s$		0,60	0,70	0,80	0,85	0,90	0,90	0,90
	$D \cdot V_S \geq 6\ m^2/s$		0,60	0,65	0,70	0,70	0,75	0,80	0,80
		b/d							
	Rectangular hollow sections over 356 mm square and 254 × 457 mm rectangular	2	1,55	1,75	1,95	2,10	2,20		
		1	1,40	1,55	1,75	1,85	1,90		
		0,5	1,0	1,20	1,30	1,35	1,40		
		0,25	0,80	0,90	0,90	1,0	1,0		
									
Single lattice frames	Flat-sided sections					1,70			
	Circular sections where:								
	$D \cdot V_s < 6\ m^2/s$					1,10			
	$D \cdot V_s \geq 6\ m^2/s$					0,80			
Machinery houses etc.	Rectangular clad structures on ground or solid base					1,10			

[1] See Fig. 2.2.4.1.4.1.

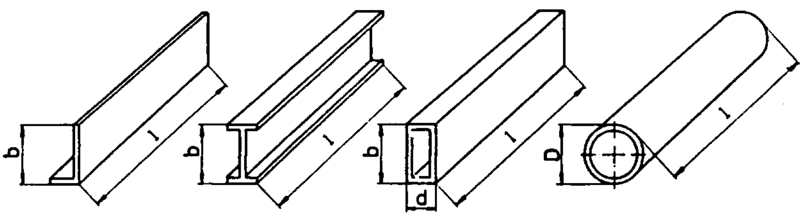
The wind load on single lattice frames may be calculated on the basis of the coefficients for the individual members given in the top part of Table T.2.2.4.1.4.1. In this case the aerodynamic slenderness of each member shall be taken into account. Alternatively the overall coefficients for lattice frames constructed of flat-sided and circular sections given in the middle part of the table may be used.

Where a lattice frame is made up of flat-sided and circular sections, or of circular sections in both flow regimes ($D \cdot V_S < 6\ m^2/s$ and $D \cdot V_S \geq 6\ m^2/s$) the appropriate shape coefficients are applied to the corresponding frontal areas.

Where gusset plates of normal size are used in welded lattice construction no allowance for the additional area presented by the plates is necessary, provided the lengths of individual members are taken between the centres of node points.

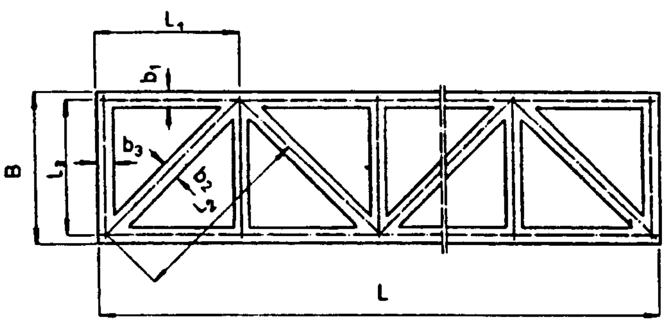
Shape coefficients obtained from wind-tunnel or full-scale tests may also be used.

(I) Aerodynamic slenderness: $\dfrac{\text{length of member}}{\text{breadth of section across wind front}} = \dfrac{l^*}{b} \text{ or } \dfrac{l^*}{D}$



* In lattice construction the lengths of individual members are taken between the centres of adjacent node points. See diagram below.

(II) Solidity ratio $= \dfrac{\text{area of solid parts}}{\text{enclosed area}} = \dfrac{A}{A_e} = \sum_{i}^{n} \dfrac{li \times bi}{L \times B}$



(III) Spacing ratio $= \dfrac{\text{distance between facing sides}}{\text{breadth of members across wind front}} = \dfrac{a}{b} \text{ or } \dfrac{a}{B}$

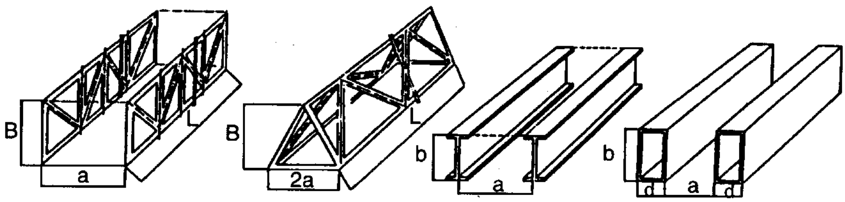


Fig. 2.2.4.1.4.1 Definitions: Aerodynamic Slenderness, Solidity Ratio, Spacing Ratio, and Section Ratio

for 'a' take the smallest possible value in the geometry of the exposed face.

(IV) Section ratio $= \dfrac{\text{breadth of section across wind front}}{\text{depth of section parallel to wind flow}} = \dfrac{b}{d}$

2.2.4.1.4.2 Multiple frames of members: shielding factors

Where parallel frames or members are positioned so that shielding takes place, the wind loads on the windward frame or member and on the unsheltered parts of those behind it are calculated using the appropriate shape coefficients. The wind on the sheltered parts is multiplied by a shielding factor η given in Table T.2.2.4.1.4.2. Values of η vary with the solidity and spacing ratios as defined in Fig. 2.2.4.1.4.1.

Table T.2.2.4.1.4.2 Shielding coefficients

Spacing ratio a/b	Solidity ratio A/A_e					
	0,1	0,2	0,3	0,4	0,5	≥ 0,6
0,5	0,75	0,40	0,32	0,21	0,15	0,10
1,0	0,92	0,75	0,59	0,43	0,25	0,10
2,0	0,95	0,80	0,63	0,50	0,33	0,20
4,0	1,0	0,88	0,76	0,66	0,55	0,45
5,0	1,0	0,95	0,88	0,81	0,75	0,68
6,0	1,0	1,0	1,0	1,0	1,0	1,0

Where a number of identical frames or members are spaced equidistantly behind each other in such a way that each frame shields those behind it, the shielding effect is assumed to increase up to the ninth frame and to remain constant thereafter.

The wind loads are calculated as follows:

On the 1st frame $F_1 = A \cdot q \cdot C_f$ in N

On the 2nd frame $F_2 = \eta \cdot A \cdot q \cdot C_f$ in N

On the nth frame (where n is from 3 to 8)	$F_n = \eta^{(n-1)} \cdot A \cdot q \cdot C_f$	in N
On the 9th and subsequent frames	$F_9 = \eta^8 \cdot A \cdot q \cdot C_f$	in N
The total wind load is thus:		
Where there are up to 9 frames	$F_{total} = [1 + \eta + \eta^2 + \eta^3 + \cdots \eta^{(n-1)}]A \cdot q \cdot C_f = A \cdot q \cdot C_f \left(\frac{1-\eta^n}{1-\eta}\right)$	in N
Where there are more than 9 frames	$F_{total} = [1 + \eta + \eta^2 + \eta^3 + \cdots \eta^8 + (n-9)\eta^8]A \cdot q \cdot C_f = A \cdot q \cdot C_f \left[\left(\frac{1-\eta^9}{1-\eta}\right) + (n-9)\eta^8\right]$	in N

Note The term η^x used in the above formula is assumed to have a lower limit of 0,10. It is taken as 0,10 whenever $\eta^x < 0{,}10$.

2.2.4.1.4.3 Lattice towers

In calculating the "face-on" wind load on square towers, in the absence of a detailed calculation, the solid area of the windward face is multiplied by the following overall force coefficient:

For towers composed of flat-sided sections	$1{,}7 \cdot (1 + \eta)$
For towers composed of circular sections	
where $D \cdot V_S < 6\ m^2/s$	$1{,}1 \cdot (1 + \eta)$
where $D \cdot V_S \geq 6\ m^2/s$	1,4

The value of η is taken from Table 2.2.4.1.4.2 for $a/b = 1$ according to the solidity ratio of the windward face.

The maximum wind load on a square tower occurs when the wind blows on to a corner. In the absence of a detailed calculation, this load can be considered as 1,2 times that developed with 'face-on' wind on one side.

2.2.4.1.4.4 Parts inclined in relation to the wind direction

Individual members, *frames*, *etc*.
Where the wind blows at an angle to the longitudinal axis of a member or to the surface of a frame, the wind load in the direction of the wind is obtained from:

$F = A \cdot q \cdot C_f \sin^2 \theta$ in N where F, A, q and C_f are as defined in Subsection 2.2.4.1.3 and θ is the angle of the wind ($\theta < 90°$) to the longitudinal axis or face.

Lattice trusses and towers
Where the wind blows at an angle to the longitudinal axis of a lattice truss or tower, the wind load in the direction of the wind is obtained from:

$$F = A \cdot q \cdot C_f \cdot K_2 \qquad \text{in N}$$

where F, A, q and C_f are as defined in Subsection 2.2.4.1.3 and

$$K_2 = \frac{\theta}{50(1{,}7 - Sp/S)}$$

which cannot be less than 0,35 or greater than 1.

Where θ is the angle of the wind in degrees ($\theta < 90°$) to the longitudinal axis of the truss or tower.

Sp is the area in m^2 of the bracing members of the truss or tower projected on to its windward plane.

S is the area in m^2 of all (bracing and main) members of the truss or tower projected on to its windward plane.

The value of K_2 is assumed to have lower and upper limits of 0,35 and 1,0 respectively. It is taken as 0,35 whenever the calculated value $<0{,}35$ and as 1,0 whenever the calculated value $> 1{,}0$.

The DIN–BS–ISO and CEN standards calculate the wind influence in a manner which deviates slightly from FEM.

In the USA and a number of other countries it is permissible to make an accurate scale model (scale 1 : 50) of the crane and to test the scale model in an approved aeronautical laboratory, e.g. in a windtunnel. The model also has to include aerodynamically representative surfaces for the walkways, stairways, ladders, platforms, and other secondary details as a minimum requirement.

The results have to be determined for the boom in the operating positions as well as in the stowed positions and with the trolley located in the critical point with and without load. Wind testing must involve wind from several different directions.

In general this could result in a lower wind force on the crane, e.g. $F = c \cdot q \cdot A$ then becomes for the full crane: $F = 1{,}4 \cdot q \cdot A$. This is somewhat lower than indicated by the standards.

A crane builder has to state openly how the windforces on the crane have been calculated and show which motor powers and motor torques this has led to. It must be kept in mind that it is a matter of *torque* which has to be delivered by the motors to the crane or to the trolley.

$$M = \frac{N \cdot 9550}{n} \text{Nm}$$

where

M = torque in Nm on the motorshaft
N = the motor power in kilowatts
n = number of revolutions per minute of the motor

A note with regard to the standards

It has been decided that the European Standards will be replaced by the CEN standards (CEN = Comité Européen de Normalisation). These CEN standards are now developed (see Sections 5.1–5.3).]

To be able to meet the wind force on the crane only it is necessary that

$$M = \frac{\Sigma(c \cdot q \cdot A) \cdot D/2}{i \cdot \eta}$$

where

M = total torque, needed on the motorshaft(s), to drive the crane against the windforce
$\Sigma(c \cdot q \cdot A)$ = total windforce
$D/2$ = radius of the driven wheel
i = reduction of the gearings between the driven wheel and the driving motor
$i = n_{\text{motor}} : n_{\text{wheel}}$
η = total efficiency of the drive mechanism

The motors have to be able to deliver the necessary torque also at low speeds during such a time that the crane driver can drive the crane against the storm towards the storm pot, where the crane can be anchored.

Considerations that must be borne in mind for a crane travelling mechanism are (see also Chapter 3):

- the resistance due to nominal travelling;
- the resistances due to the influence of the wind;
- the resistances due to the acceleration of the linear moving masses; and
- the resistances due to the acceleration of the rotating masses.

The wind speeds and aerodynamic pressures become greater at greater heights from the ground.

Table 1.5.2 Windspeeds etc., at greater heights

Height above quay in metres	*Storm windspeed in m/sec*	*Aerodynamic pressure*	
		q in N/m²	*(in lbs/sq ft)*
0–20 metres	36	800	(16)
20–100 metres	42	1100	(22)
> 100 metres	46	1300	(26)

1.6 Hatchless container vessels

Container vessels with hatches are well known. The deck-containers are then loaded on top of the hatches and must be very well lashed with lashings and semi-automated twist locks (SATLs) to prevent containers being lost during a storm etc.

Unlashing respective. Lashing the containers, handling the SATLs, and opening and closing the hull of vessels with hatches takes a considerable time.

In order to overcome these problems, naval architects designed the so-called 'hatchless vessels'. These vessels have no hatches and reinforced cell-guides run from the bottom of the vessel, up to almost the top of the highest deck-container. Lashing of these deck-containers and handling the hatches is then no longer necessary. This hatchless construction saves a lot of work and time during unloading or loading the vessel; in addition, the danger of losing containers in a storm is fully avoided.

Nedlloyd and other stevedores have built a number of these hatchless vessels. Nedlloyd states that they achieved 10–15 percent greater productivity, although now all containers have to be hoisted over the higher cell-guides.

The well-known Dutch naval architect Ir. E. Vossnack has made designs for a series of modernized hatchless vessels which show interesting design details. Figures 1.6.1 and 1.6.2 show some of the design sketches of Ir. E. Vossnack.

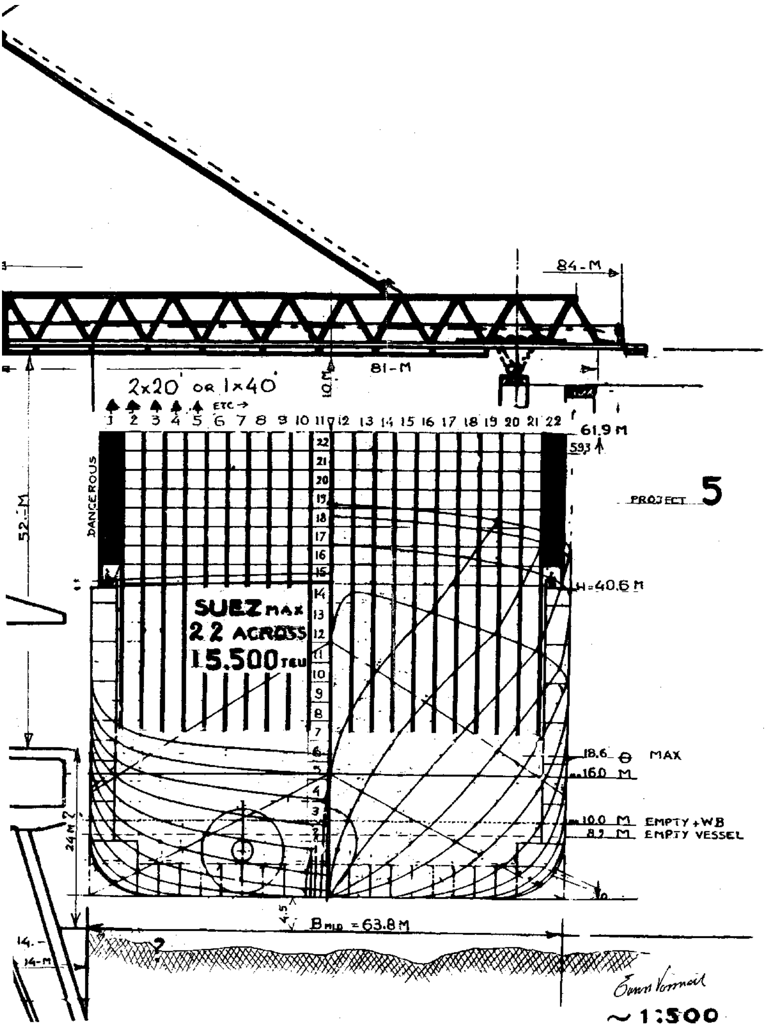


Fig. 1.6.1 Hatchless vessel, 22 across

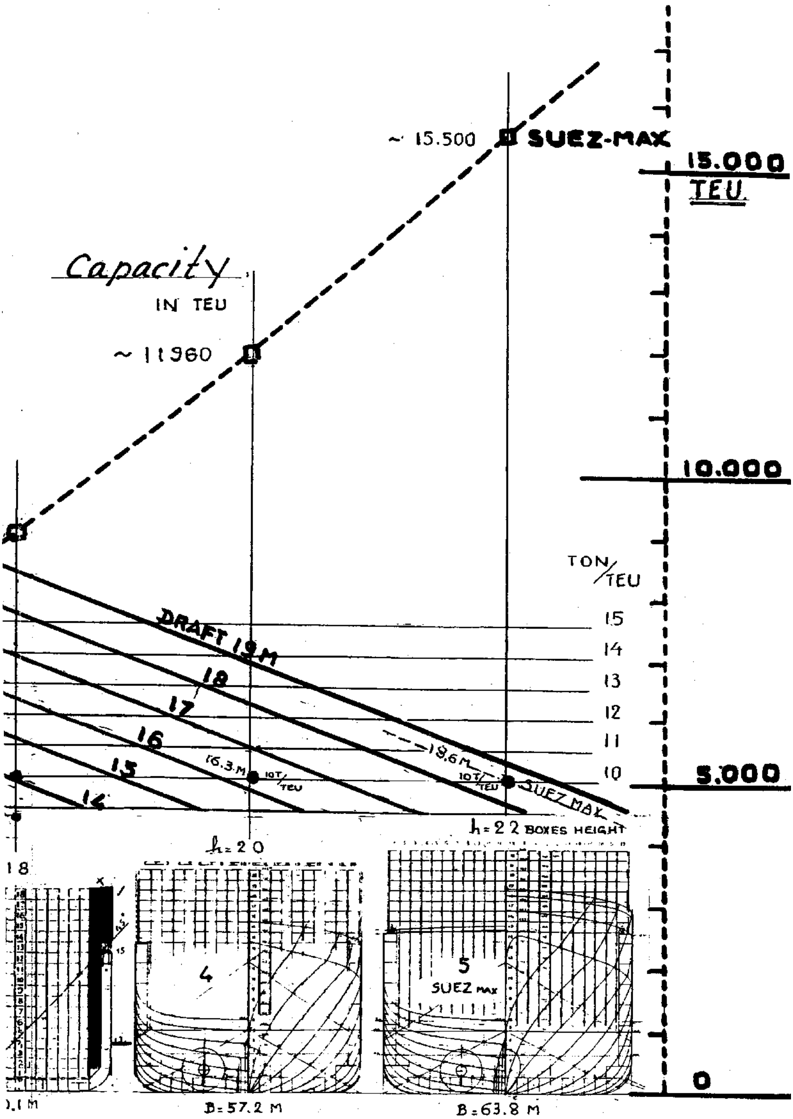


Fig. 1.6.2 Capacity, width and draft, etc.

Fig. 1.6.3 Hatchless vessel

Chapter 2

Wire Ropes

2.1 Wire rope reeving systems

The system of the wire rope reeving differs from one crane type to another. Two of the main reeving systems are indicated in the following figures. In container cranes, the safety of the wire rope against rupture

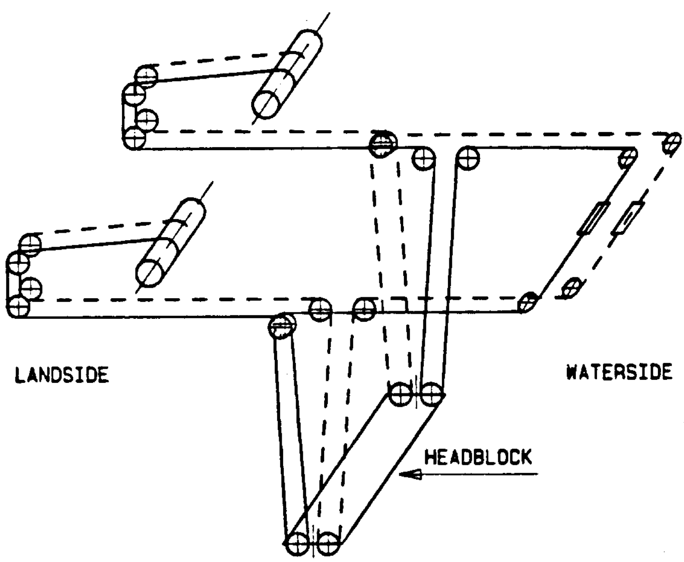


Fig. 2.1.1 Normal hoist wire rope scheme for a container crane

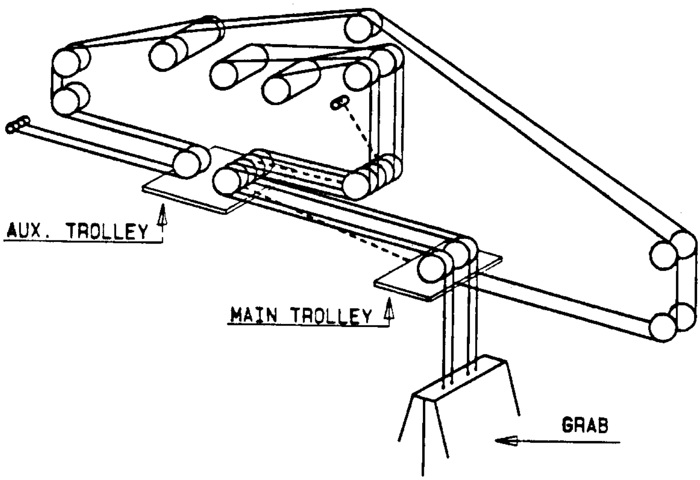


Fig. 2.1.2 Wire rope scheme for grab-unloader with main and auxiliary trolley

should be a factor of six. For grab-unloaders the system in Fig. 2.1.2 with a main trolley and an auxiliary trolley is very popular. However, other systems should be mentioned, such as:

- machinery trolleys with complete hoisting winches on the trolley;
- the 'fleet through' reeving system;
- the 'in bight of line' system.

In heavy duty grab-unloaders the safety of the wire ropes against rupture should be approximately eight.

In other crane types many different wire rope reeving systems are used, each system suitable for the special duty. Examples are:

- stacking cranes;
- laddle cranes;
- overhead travelling cranes;
- tower cranes;
- slewing cranes for general cargo – and/or grabbing duties;
- level-luffing cranes for general cargo – and/or grabbing duties;
- off-shore cranes.

2.2 Influencing the lifetime of wire ropes

The main points that influence the lifetime, wear and tear of wire ropes are:

- the rope reeving system;

- the chosen type of wire ropes;
- the diameter of the drums and sheaves;
- the influence of reversed bendings;
- the distance between the cable sheaves;
- the speeds with which the wire ropes run through the sheaves;
- the ratio between normal working load and the maximum load in the wire ropes;
- the safety factor; being the ratio between the breaking load and the normal working load;
- the choice of the hardness in the groove of the sheaves;
- the deviation, or fleet-angle between the wire rope and the sheave, respectively between the drum groove and wire rope;
- the greasing or lubrication of the wire ropes and the frequency of greasing or lubrication;
- the way in which the wire ropes might run through dirt, ore residues, abrasive materials, etc.;
- mechanical damage through hitting cell-guides or coamings in the ships;
- internal and external corrosion.

2.3 Drum diameters and wire rope sheave diameters

Almost every country has its own standards for these diameters (D) in relation to the rope diameter (d). Some standards indicate that the wire rope sheave diameter should be increased when the speed of the wire rope, running through the sheave, is more than $V = 4$ m/sec. Then

$$Dv = (V:4)D.$$

A general warning is that the drum diameter and wire rope sheave should not be made too small in diameter.

For faster cranes:

$$D/d = 30.$$

For fast unloaders:

$$D/d = 36.$$

Where D = wire rope sheave diameter or drum diameter, centre to centre of wire rope, and d = wire rope diameter.

2.4 The choice of wire ropes

Without going into too much detail, it can be stated that the choice is mainly between ordinary (regular) or cross lay wire ropes and Lang's lay wire ropes. Both types should preferably be used with a steel core. They should be galvanized and always fully lubricated or greased. The tensile strength should be approximately 1770 N/mm^2.

The safety factor, being the ratio between the minumum breaking load and the normal working load, must be in accordance with the national standards. Wire ropes, such as the well-known Casar wire ropes, the Diepa wire ropes, the Bridon–Dyform wire ropes, and others can, under certain circumstances, give a longer lifetime than cross lay and Lang's lay wire ropes. This can normally only be proven by trials on the cranes; the outcome of which is difficult to predict.

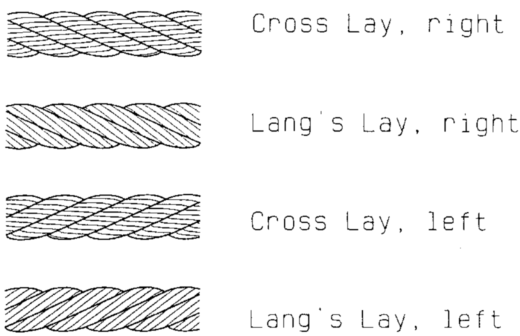


Fig. 2.4.1 Wire rope types

2.5 Fleet angles; grooves on drums and in wire rope sheaves

Deviation into the direction of the groove

(a) Deviation by the groove $L2 = (L:\pi \cdot D) \cdot S$
Maximum deviation $L1$
Fleet angle between the drum and the sheave $V1 = (L1 - L2):L = 1:\ldots.$

(b) Fleet angle between the wire rope and the sheave $V2 = L1:L = 1:\ldots.$

Deviation against the direction of the groove

(c) Deviation against the direction of the groove $L2 = (L:\pi \cdot D) \cdot S$
Maximum deviation $L3$
Fleet angle between the drum and the sheave $V3 = (L3 + L2):L = 1:\ldots.$

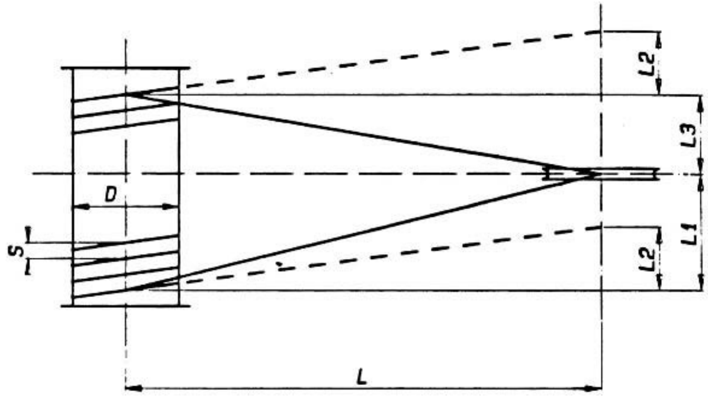


Fig. 2.5.1 Fleet angles on drums and sheaves

(d) Fleet angle between the wire rope and the sheave $V4 = L3:L = 1:\ldots.$

(Where S = the pitch and D = the drum diameter.)
$V1$ and $V3$ should be approximately 1:20.
$V2$ and $V4$ should be at least 1:16, but normally approximately 1:20.

(When degrees are measured it is preferably that $L1:L$ and $L3:L$ should not result in an angle of more than 2,5 degrees; the preferred maximum is 2 degrees.)

It is better to have the sheave in such a position that the maximum deviation *into* the direction of the groove, and the maximum deviation *against* the direction of the groove are equal, and preferably 1:20.

Controlling the drum grooves

In order to control whether the wire rope is bending off too sharply over the rim of the groove on the drum, when running off into the direction of the groove, and when it is running off from the drum against the direction of the groove, a drawing should be made when $V1$ and $V3$ have lower figures.

Professor Ernst indicated figures in his book *Die Hebezeuge* and the Belgian standards NBN–E 52-004 (1980) give the following useful diagrams (see Figs 2.5.2–2.5.6).

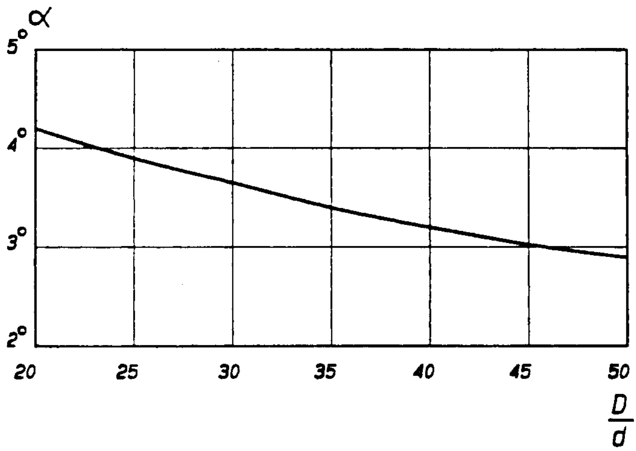


Fig. 2.5.2 Wire rope running off from the drum; maximum allowable deviation into the direction of the groove, measured from the tangent of the groove

***D*=drum diameter, centre to centre of wire rope**
***d*=wire rope diameter**

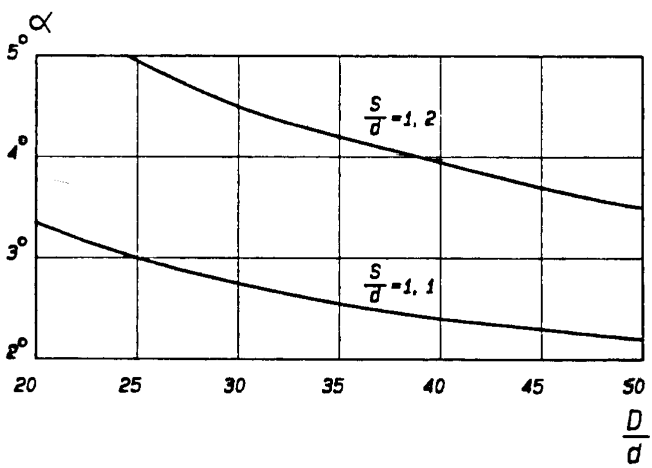


Fig. 2.5.3 Wire rope running off from the drum; maximum allowable deviation against the direction of the groove, measured from the tangent of the groove

***D*=drum diameter, centre to centre of wire rope**
***d*=wire rope diameter**
***S*=pitch**

The grooves on drums and in sheaves

On the drums the grooves must be deep enough to guide the wire ropes correctly.

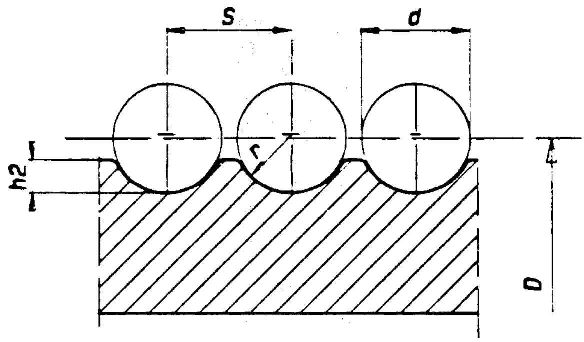


Fig. 2.5.4 Grooves on drums

$h_2 \simeq 0{,}3d$
$r \simeq 1{,}05 \cdot (d:2)$
$S_{\min} \simeq 1{,}1d$

Drum material = Fe510 (S355)

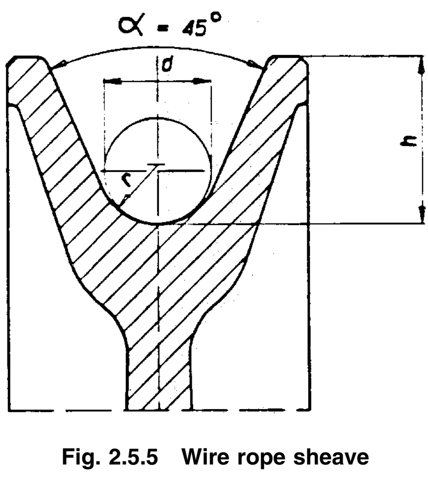


Fig. 2.5.5 Wire rope sheave

$h = 1{,}5d$ to $2d$
$r \simeq 1{,}05(d:2)$

Hardness of the material in the groove = 260 to 350 HBr.

Occasionally wire rope sheaves, made of a type of nylon are chosen. Strangely enough the use of only nylon wire rope sheaves can lead to unexpected, sudden wire rope breakage.

Note: The CEN standards will mention $\alpha = 52$ degrees instead of $\alpha =$ 45 degrees.

Controlling the wire rope sheave opening

This is diagrammatically displayed in Fig. 2.5.6.

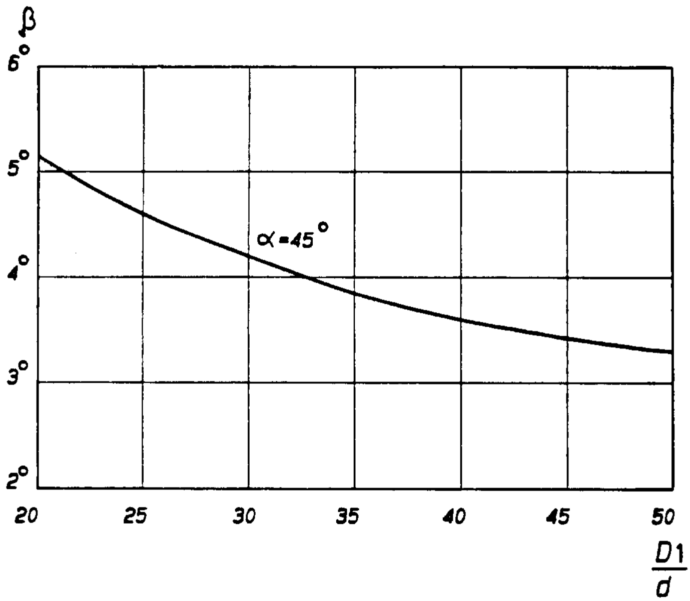


Fig. 2.5.6 Wire rope running off from the wire rope sheave. Maximum allowable fleet angle measured from the tangent of the groove (only for the controlling of the sheave opening)

D_1 =wire rope sheave diameter, centre to centre of wire rope
d=wire rope diameter
α=45 degrees (see Fig. 2.5.5)

2.6 The bending angle over sheaves; the ultimate tensile limit

The bending angle over a sheave should be at least some 60 degrees and should cover preferably at least 1,5 × the lay lengths in order to avoid shortening the wire rope's lifetime. Another important feature is the ultimate tensile limit. The ultimate tensile limit is approximately 50 percent of the minimum breaking load of the wire rope.

Through occasional heavy shocks, a miscalculation, or added tensioning of the wire ropes by a tensioning winch, the wire rope can occasionally be overloaded. Then the load in the wire rope could reach the ultimate tensile limit. If this occurs, the wire rope may then have a very short lifetime and may even rupture unexpectedly.

2.7 The lifetime of wire ropes

In Section 2.2 the main points that influence the lifetime and the wear and tear of wire ropes were listed. Occasionally the lifetime of a wire rope is extremely short because of mechanical damage from hitting the cell-guides or coamings in a ship. Table 2.7.1 lists some lifetime ideas for the wire ropes in container quay cranes. They can be considered as averages.

It has to be said that the lifetime of the very fast running wire ropes in Lang's lay construction was, in general, no better than that of the ordinary or cross lay wire ropes.

Table 2.7.1 Lifetime of wire ropes

		Lifetime in moves	
Client	*Case*	*Hoisting wire ropes*	*Trolley travelling wire ropes*
P	A	140 000	64 000
	B	120 000	100 000
	C	84 000	120 000
	D	55 000	46 000
Q	K	175 000	120 000
	L	140 000	106 000
	M	110 000	138 000
R		After 100 000 moves the wire ropes are changed	After 100 000 moves the wire ropes are changed
S		80 000	?
T		64 000	?
U		250 000	Machinery trolley

Note: The boom hoist wire ropes are normally changed once every five years or sometimes even once every ten years.

Pressures between the wire rope and sheaves or drums

The tensile loads and the bending loads are the main causes for fatigue, although mechanical damage is often the main reason why a wire rope should be changed.

If it is assumed that the wire rope is running in and over a well-fitted groove; the pressure between the wire rope and the groove is given by

$$P = \frac{F}{D/2 \cdot d} \text{ N/mm}^2$$

where

P = pressure in the groove (N/mm^2)
F = the wire rope force (N)
$D/2$ = the radius of the sheave or drum (mm)
d = the wire rope diameter (mm)

The maximum permitted pressure P_{max} (N/mm^2) is

– on steel Fe 510 (S 355) approximately 7,0 N/mm^2
– on manganese or alloyed steels approximately 20,0 N/mm^2

Maintenance

Wire ropes and wire rope sheaves should be regularly inspected. Greasing is most important even though the wire ropes have already been greased, externally and internally during fabrication. If the internal greasing has not been performed correctly, the lifetime of a wire rope is dramatically reduced.

2.8 Wire rope strength

Although there are many types of construction of wire ropes, only a summary of the minimum Breaking Load is given for one type of wire rope with a tensile strength of 1770 N/mm^2 and a steel core.

Table 2.8.1 Wire rope strength

Wire rope diameter (mm)	*Minimum breaking load* (kN)	*Tensile strength* (N/MM^2)	*Weight per 100 m* (kN)
20	250	1770	1,59
22	305	1770	1,93
24	363	1770	2,29
26	425	1770	2,69
28	494	1770	3,12
30	567	1770	3,58
32	645	1770	4,08
34	728	1770	4,60
36	817	1770	5,16

Note: The wire rope hardness of such a wire rope is between 400 and 500 HBr.

Chapter 3

Drives; Calculating Motor Powers

3.1 Driving systems

Electricity is the normal power used for driving cranes, so this is the main system considered here. The types of drives used for the crane mechanisms are given below.

The squirrel cage motor with fluid coupling

In simple cases, it can be worthwhile to consider the alternating current squirrel cage motor, with a fluid coupling as the driving element for the horizontal movements. However, there is then no possibility for real speed regulation. This can only be executed by 'inching' or small and repeated movements, commanded by the push button knob or the toggle-switch. For driving a belt conveyor, the squirrel cage motor with

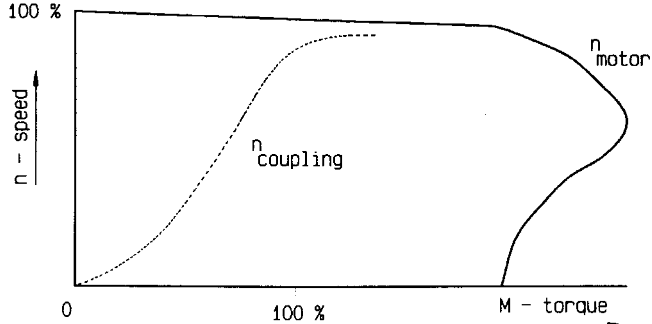


Fig. 3.1.1 Squirrel cage motor with fluid coupling

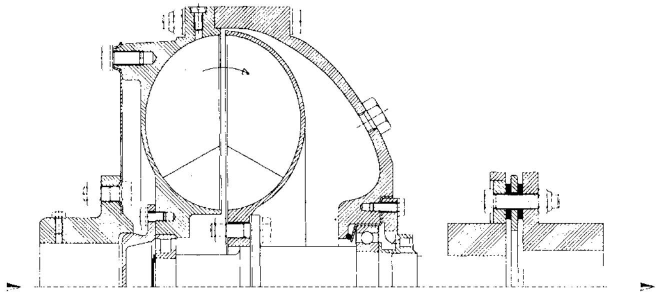


Fig. 3.1.2 Fluid coupling

fluid coupling is an excellent type of drive as it gives smooth acceleration of the complete belt system.

The slipring motor

The slipring motor is a drive which is now little used but it is still worth mentioning. The alternating current slipring motor is speed-controlled by resistances. These resistance-steps can be switched on or off by the controller. If torque is required: the more resistance, the lower the speed. 'No resistance' gives the speed curve of the normal squirrel cage motor. The brushes of the motor need regular maintenance; the resistances can burn out and rust. Therefore resistances made of stainless steel have preference.

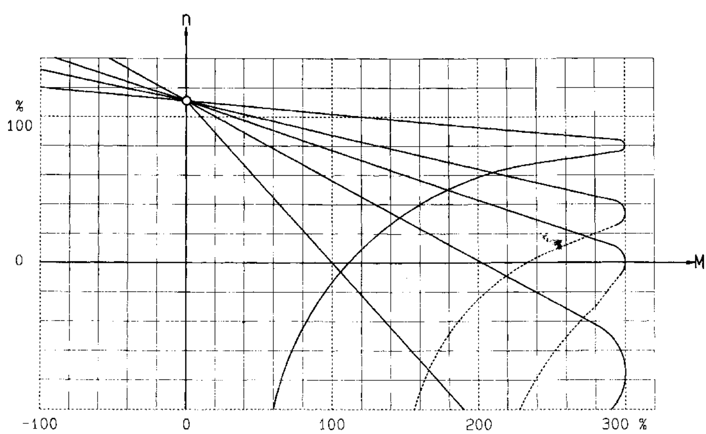


Fig. 3.1.3 Slipring motor: resistance controlled

The Ward–Leonard drive

The Ward–Leonard (WL) drive can be considered as a 'better DC drive'. (The DC drive with resistance control is not further described.) The more complicated WL drive has great advantages compared to drives with slipring motors or DC motors with resistance control.

The main motor, which is a squirrel cage motor, runs at a constant speed during the workshift on the crane. It drives a Ward–Leonard generator for each mechanism. The generator is directly coupled to the main motor and gives a regulated voltage and current to the respective motor which forms the drive-element of the crane mechanism. The speed control of this drive-element can be stepless.

With a three-field generator like the Ward–Leonard–Krämer the maximum torque can be fixed exactly at the desired level. This gives excellent drives for the hoisting mechanisms of grabbing cranes which dredge under water and for the drives of cutter-dredgers and similar devices. Cosphi compensation is not necessary. The Ward–Leonard–

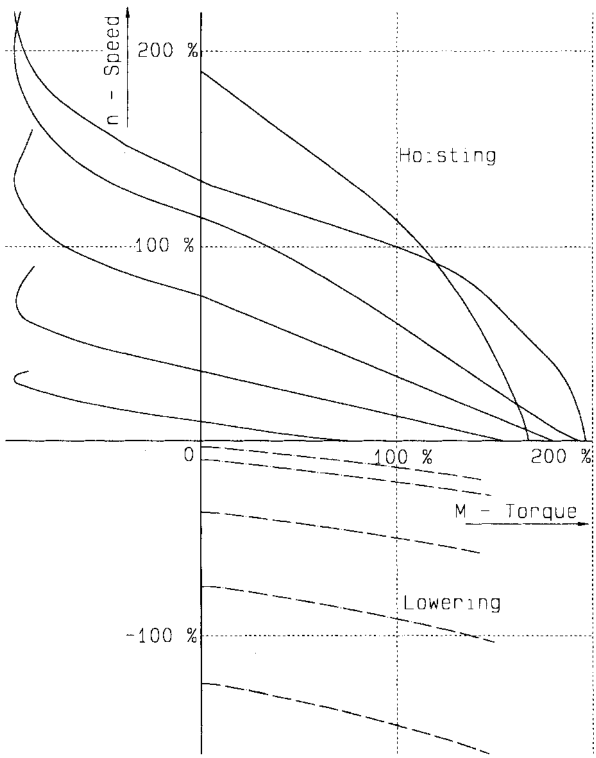


Fig. 3.1.4 Ward–Leonard–Krämer (hoist motion)

Krämer drive has advantages when the current-supply delivery net is weak or when the main drive element is a diesel engine. A factor, which must be carefully monitored, is the average accelerating torque. Knowledge of how to design and manufacture these powerful Ward–Leonard drives has unfortunately been largely lost.

Direct current full-thyristor systems

In the last twenty years the direct current full-thyristor drive has become the successor to the resistance-controlled AC drives and DC drives and the Ward–Leonard drives.

The stepless controlled full-thyristor direct current motor is available for all mechanisms and all capacities. It can be regarded as fool proof. Regular maintenance is needed to attend to the brushes, and collectors in the motors. Dust caused by wear and tear of the brushes has to be removed from time-to-time and the brushes have to be adjusted, checked, and replaced to prevent breakdown and loss of efficiency. These motors can be totally enclosed or drip-watertight, self-ventilated or ventilated by an external, continously running ventilator (force-ventilated). Field weakening can occur, normally to a level of approximately 1500 to 2000 rev/min depending on the power range and field compensation. The normal voltage is 400 V or 500 V. Cosphi compensation is needed to achieve a cosphi of approximately 0,9.

Alternating current drives with frequency control

To reduce maintenance on the motors as much as possible, the manufacturers of electrical systems have developed and now use AC motors with frequency control. Since 1995 a good working system has been achieved. AC frequency control is also available for hoisting mechanisms using large amounts of power.

The motors are of a simple design. However these are special squirrel cage motors. The electrical control is somewhat more complicated than that of the full-thyristor systems, and forced ventilation is not normally required. Control of these motors is always stepless. Field weakening, up to 2000 to 2200 rev/min – based on a four-pole motor, is possible by increasing the frequency. Torque–speed curves can be adjusted within a limited range.

It is safe to assume that the research and development of the design of motors will continue and that further advances will be made. However, this drive offers the most appropriate and suitable answer for the next ten years. Cosphi compensation may be necessary to achieve a cosphi level of approximately 0,9 depending on the type of the drive.

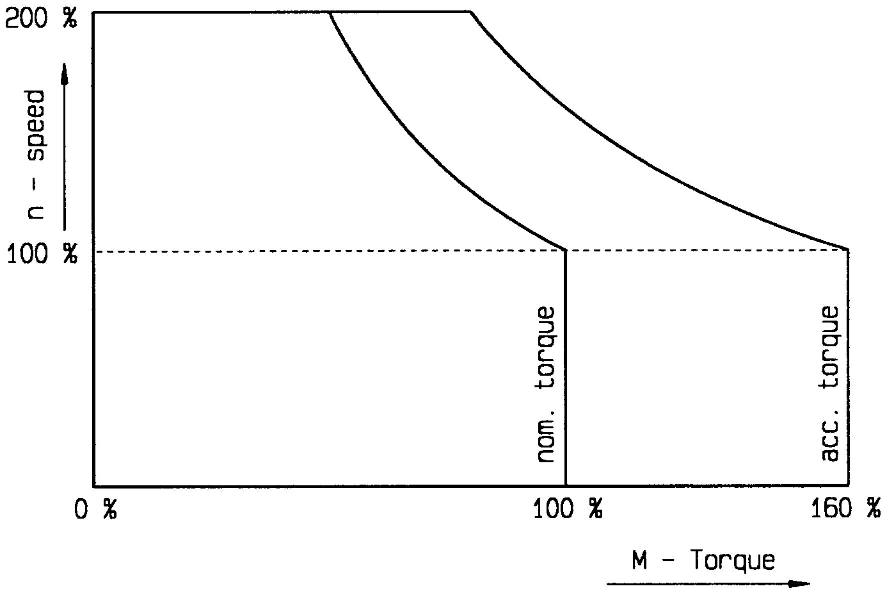


Fig. 3.1.5 DC full thyristor

In low speed crane-travelling mechanisms, the option of using one drive for all the motors under the two sill-beams of the cranes is possible. Because all the motors will receive the same frequency, synchronization between the motors is not absolutely necessary providing that the wheel loads and the wind loads on each sill-beam of the crane do not differ significantly. However, it is preferable to use one drive for each sill-beam and also to make 'cross-over' connections between the motors on the two sill-beams. This ensures exact synchronization.

Warning Especially with AC frequency control, but often also with DC-Full-Thyristor Control the Electromagnetic Compatability (EMC) due to the Higher Harmonics plays an important role.

To prevent disturbances by this Electro Magnetic Interference (EMI) special double-shielded cables must be used. These screens or shields consist of a copper foil wrapping and optimized copper wire braiding.

On both ends of the cable special EMC glands must be used. These must be well-earthed and connected to steel boxes.

In the bigger motors insulated bearings should also be used.

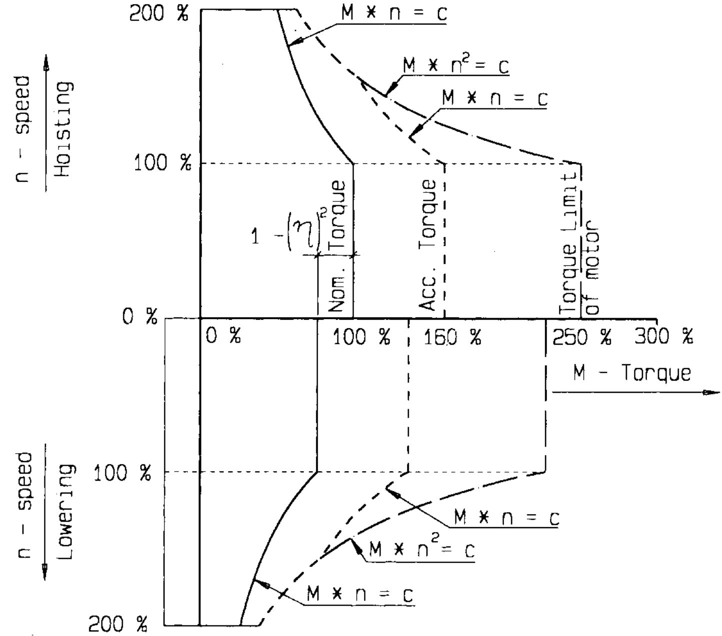


Fig. 3.1.6 AC frequency control: torque–speed diagram for hoisting/lowering

Fig. 3.1.7 2 × 800 kW Holec AC frequency control motors in the hoisting winch of a grab-unloader

Hydraulic drives

We now concentrate on the Hägglunds hydraulic drive, which consists of a control system; an electric motor; an oil tank; a pump; and a hydraulic motor. The pump is driven by an electric motor, which runs with a fixed speed. The oil flow from the pump is controlled by either a Squashplate or a tilting cylinder block, the angle of which can be changed by a signal from the control system. The motor pumps the oil which flows into the motor cylinders and presses the pistons radially out towards the camring. The speed of the motor is stepless variable. This system has a low moment of inertia and a high starting torque (200 to 300 percent of a nominal rated torque). A brake system can also be provided on these drives.

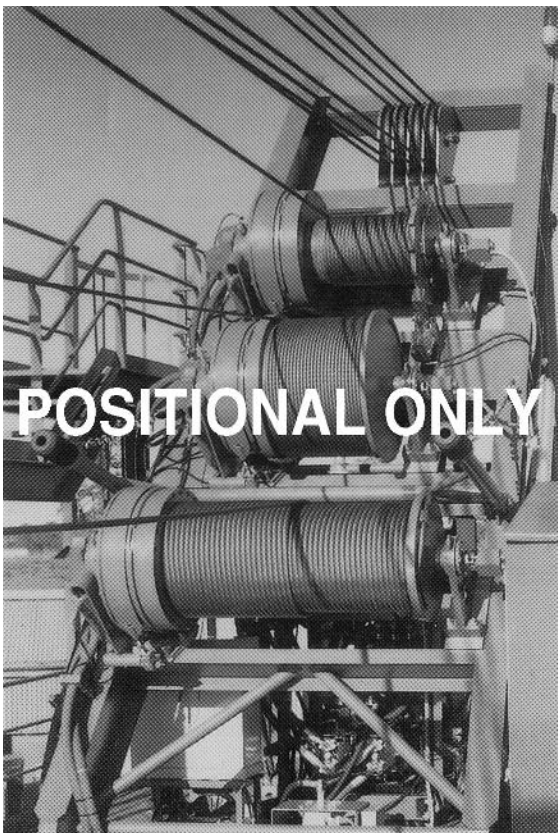


Fig. 3.1.8 Winches with Hägglunds hydraulic drives

3.2 Numbers of wire rope sheaves in the hoisting mechanisms of different reeving systems

As already mentioned in Wire Rope Reeving Systems (Section 2.1), there are quite a number of reeving systems for hoisting mechanisms. The main types are considered in Figs 3.2.1(a) to 3.2.1(e).

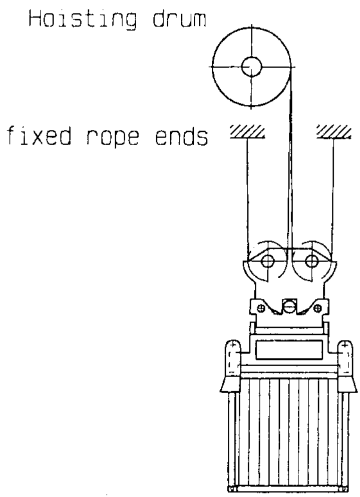


Fig. 3.2.1(a) Container cranes with machinery trolley. (Hoisting winch on the trolley.) Number of rope sheaves: minimum

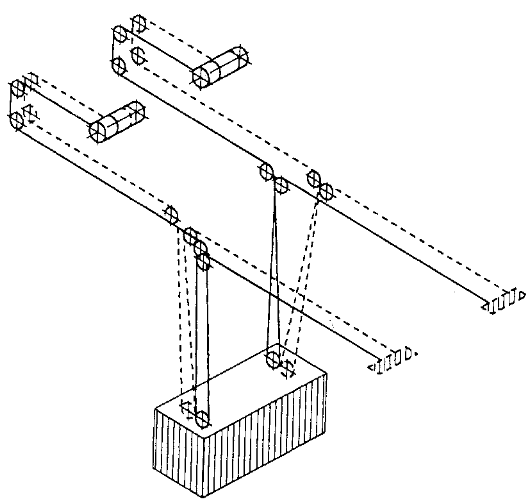


Fig. 3.2.1(b) Container cranes with rope trolley. (Hoisting winch fixed on the bridge.) Number of rope sheaves: depending on wire rope layout

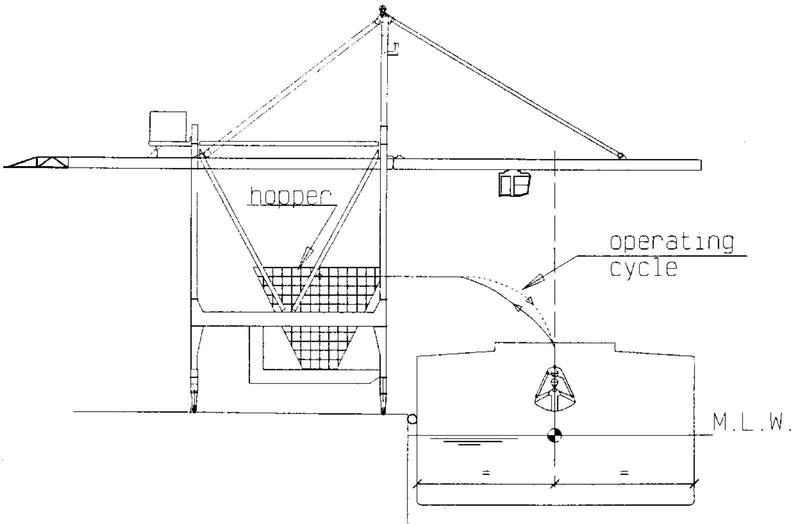


Fig. 3.2.1(c) Grab unloader with main and auxiliary trolley. Number of rope sheaves: see Fig. 2.1.2

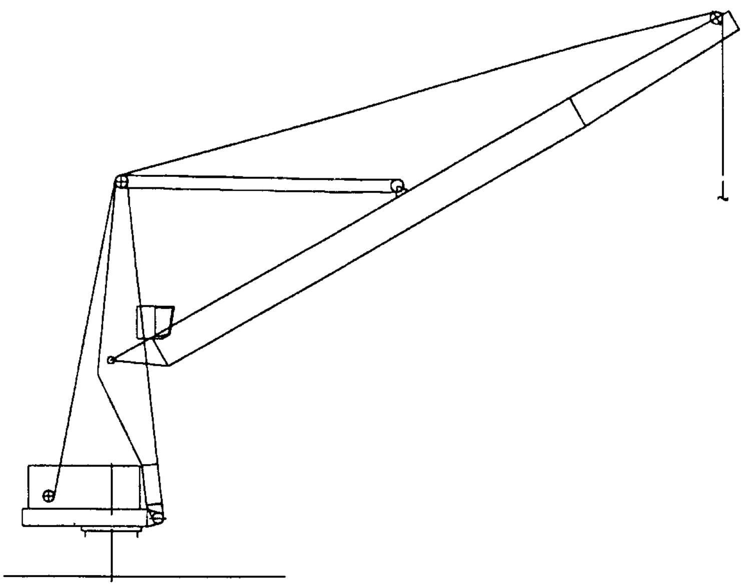


Fig. 3.2.1(d) Level luffing crane. Number of rope sheaves: see sketch

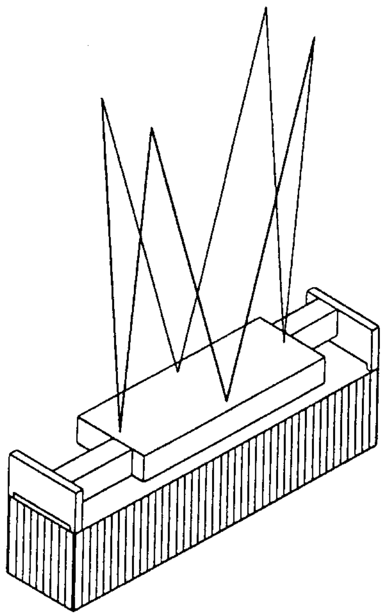


Fig. 3.2.1(e) Stacking crane with 'rope tower'. Number of sheaves: depends on rope system in trolley

3.3 Calculating the requisite power of the hoisting motors

For calculating the requisite motor power the following items must be considered:

(a) the resistance due to normal (nominal) hoisting;
(b) the resistance due to acceleration of the rotating masses;
(c) the resistance due to acceleration of the linear moving masses;
(d) for the hoisting mechanism shown in Fig. 3.2.1(e) the influence of the angles α have to be taken into account, as the forces and the motor power are multiplied in this wire rope system

$$\text{with } f = \frac{1}{\cos \alpha}$$

where α is then half of the biggest angle between the wire ropes when the load is in the highest position.

Example 1

The example, shown here, is related to a container crane with a rope trolley, as in Fig. 3.2.1(b).

Main Characteristics		Example
Weight of load: spreader and container or grab and contents or hook and load		
Q	kg	$Q = 66\,000$ kg
	kN	$Q = 660$ kN
Maximum speed of the load:		
v	m/min	$v = 60$ m/min
	m/sec	$v = 1$ m/sec
Efficiency of all gearings and rope sheaves:	η_t	$\eta_t = 0{,}90$
Motor speed:	rev/min	$n = 783$ rev/min
Inertia moment on motorshaft from motor(s); break sheave(s); and gearbox:		
$J_{\text{rot}} = J_{\text{m}} + J_{\text{b}} + J_{\text{gb}}$	kg m^2	$J_{\text{rot}} = 24 + 16 + 6 = 46$ kg m^2
Acceleration time:	sec	$t_{\text{a}} = 2$ sec
Acceleration of the mass Q:		
$a = \dfrac{v}{t}$	m/sec^2	$a = \dfrac{1}{2} = 0{,}5$ m/sec^2

		Torque (Nm)	kiloWatts (kW)
1. Resistance due to nominal hoisting (full load at maximum speed):			
$N_1 = \dfrac{Q \cdot v}{\eta}$	kW		$N_1 = \dfrac{660 \cdot 1}{0{,}9} = 773\,\text{kW}$
$M_1 = \dfrac{N_1 \cdot 9550}{n}$	Nm	$M_1 = \dfrac{773 \cdot 9550}{783} = 8940\,\text{Nm}$	
2. Resistance due to accelerating the rotating masses:			
$\omega = \dfrac{n \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{783 \cdot 2 \cdot \pi}{60} = 81{,}95\,\text{rad/sec}$	
$M_2 = \dfrac{J_{\text{rot}} \cdot \omega}{t_{\text{a}}}$	Nm	$M_2 = \dfrac{46 \cdot 81{,}95}{2} = 1885\,\text{Nm}$	
$N_2 = \dfrac{n \cdot M_2}{9550}$	kW		$N_2 = \dfrac{783 \cdot 1885}{9550} = 154{,}5\,\text{kW}$
3. Resistance due to accelerating the linear masses:			
$F_3 = \dfrac{Q \cdot v}{g \cdot t_{\text{a}}}$	kN		$F_3 = \dfrac{660 \cdot 1}{9{,}81 \cdot 2} = 33{,}6\,\text{kN}$
$N_3 = \dfrac{F_3 \cdot v}{\eta}$	kW		$N_3 = \dfrac{33{,}6 \cdot 1}{0{,}9}\,\text{kW} = 37{,}33\,\text{kW}$
$M_3 = \dfrac{N_3 \cdot 9550}{n}$	Nm	$M_3 = \dfrac{37{,}33 \cdot 9550}{783} = 455\,\text{Nm}$	

		Torque (Nm)	kiloWatts (kW)
	Addition:		
1.	Nominal hoisting	$M_1 = 8940$ Nm	$N_1 = 733$ kW
2.	Acceleration of the rotating masses	$M_2 = 1885$ Nm	$N_2 = 154{,}5$ kW
3.	Acceleration of the linear moving masses	$M_3 = \underline{455 \text{ Nm}}$	$N_3 = \underline{37{,}33 \text{ kW}}$
	Total	$\Sigma M = 11\,280$ Nm	$\Sigma N = 924{,}83$ kW

During acceleration, the motor(s) can deliver more torque for a restricted time. This can vary from about 140 percent to as much as 250 percent.

For example: $f_a = 160$ percent $= 1{,}6$

The motor(s) must be able to deliver

$$N_1 = 733 \text{ kW}$$

resp.

$$\frac{\Sigma N}{f_a} \leq N_1$$

$$\frac{\Sigma N}{f_a} = \frac{924{,}83}{1{,}6} = 578 \text{ kW}$$

(which is lower than N_1: so use $N_1 = 733$ kW)

Take motor(s): $N = 733$ kW $(2 \cdot 366$ kW)

$n = 783$ rev/min

$S_3 - 60$ percent rating (see under Section 3.7)

$f_a = 160$ percent

Field weakening

Let us assume that it is also necessary to hoist (and lower) the load of 200 kN with a speed of $v = 120$ m/min ($v = 2$ m/sec) and an acceleration time of 4 sec. The motor(s) then run at $n = 2 \cdot 783 = 1566$ rev/min.

		Torque (Nm)	kiloWatts (kW)
1. Resistance due to nominal hoisting:			
$N_1 = \dfrac{Q \cdot v}{\eta_t}$	kW		$N_1 = \dfrac{200 \cdot 2}{0{,}9} = 444$ kW
$M_1 = \dfrac{N \cdot 9550}{n}$	Nm	$M_1 = \dfrac{444 \cdot 9550}{1566} = 2707$ Nm	
2. Resistance due to accelerating the rotating masses:			
$\omega = \dfrac{n \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{1566 \cdot 2 \cdot \pi}{60} = 163{,}9$ rad/sec	
$M_2 = \dfrac{J_{rot} \cdot \omega}{t_a}$		$M_2 = \dfrac{46 \cdot 163{,}9}{4} = 1885$ Nm	
$N_2 = \dfrac{n \cdot M_2}{9550}$	kW		$N_2 = \dfrac{1566 \cdot 1885}{9550} = 309$ kW
3. Resistance due to accelerating the linear masses:			
$F_3 = \dfrac{Q \cdot v}{g \cdot t_a}$	kN		$F_3 = \dfrac{200 \cdot 2}{9{,}81 \cdot 4} = 10{,}19$ kN
$N_3 = \dfrac{F_3 \cdot v}{\eta_t}$	kW		$N_3 = \dfrac{10{,}19 \cdot 2}{0{,}9} = 22{,}6$ kW
$M_3 = \dfrac{N_3 \cdot 9550}{n}$	Nm	$M_3 = \dfrac{22{,}6 \cdot 9550}{1566} = 138$ Nm	

		Torque (Nm)	kiloWatts (kW)
	Addition:		
1.	Nominal hoisting	$M_1 = 2707$ Nm	$N_1 = 444$ kW
2.	Acceleration of the rotating masses	$M_2 = 1885$ Nm	$N_2 = 309$ kW
3.	Acceleration of the linear moving masses	$M_3 = \underline{138\text{ Nm}}$	$N_3 = \underline{22{,}6\text{ kW}}$
	Total	$\Sigma M = 4370$ Nm	$\Sigma N = 775{,}6$ kW

Take motors: $N = 733$ kW
$(2 \cdot 366$ kW$)$

$n = 783/1566$ rev/min

$S_3 - 60$ percent rating (see Section 3.7)

$f_a = 160$ percent

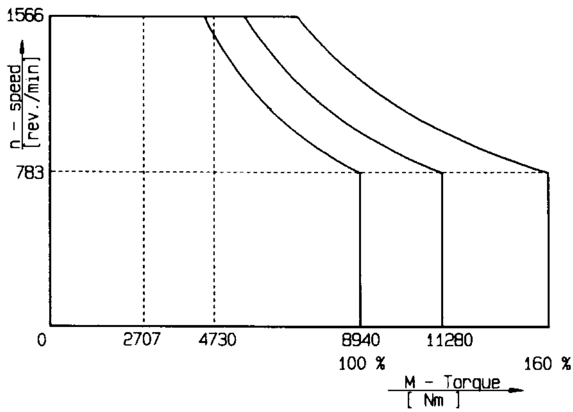


Fig. 3.3.1 DC FT torque–speed diagram

For grabbing winches

Follow the same calculation method.

Example 2

Assume the weight of the loaded grab is 36 t; the nominal speed 120 m/min.
Unloader type as Figs 2.1.2 and 3.2.1(c).
With empty grab the speed is 150 m/min.

		Torque (Nm)	kiloWatts (kW)
1. Resistance due to nominal hoisting (full load at maximum speed):			
$N_1 = \dfrac{Q \cdot v}{\eta}$	kW		$N_1 = \dfrac{360 \cdot 2}{0{,}85}$ $= 847\,\text{kW}$
$M_1 = \dfrac{N_1 \cdot 9550}{n}$	Nm	$M_1 = \dfrac{847 \cdot 9550}{783}$ $= 10\,330\,\text{Nm}$	
2. Resistance due to accelerating the rotating masses:			
$\omega = \dfrac{n \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{783 \cdot 2 \cdot \pi}{60}$ $= 81{,}95\,\text{rad/sec}$	
$M_2 = \dfrac{J_{\text{rot}} \cdot \omega}{t_{\text{a}}}$	Nm	$M_2 = \dfrac{46 \cdot 81{,}95}{2}$ $= 1885\,\text{Nm}$	
$N_2 = \dfrac{n \cdot M_2}{9550}$	kW		$N_2 = \dfrac{783 \cdot 1885}{9550}$ $= 154{,}5\,\text{kW}$
3. Resistance due to accelerating the linear masses:			
$F_3 = \dfrac{Q \cdot v}{g \cdot t_{\text{a}}}$	kN		$F_3 = \dfrac{360 \cdot 2}{9{,}81 \cdot 2}$ $= 36{,}7\,\text{kN}$
$N_3 = \dfrac{F_3 \cdot v}{\eta}$	kW		$N_3 = \dfrac{36{,}7 \cdot 2}{0{,}85}$ $= 86{,}35\,\text{kW}$
$M_3 = \dfrac{N_3 \cdot 9550}{n}$	Nm	$M_3 = \dfrac{86{,}35 \cdot 9550}{783}$ $= 1053\,\text{Nm}$	

		Torque (Nm)	kiloWatts (kW)
	Addition:		
1.	Nominal hoisting	$M_1 = 10330$ Nm	$N_1 = 847$ kW
2.	Accelerating of the rotating masses	$M_2 = 1885$ Nm	$N_2 = 154{,}5$ kW
3.	Accelerating the linear moving masses	$M_3 = \underline{1053\text{ Nm}}$	$N_3 = \underline{86{,}4\text{ kW}}$
	Total	$\Sigma M = 13\,268$ Nm	$\Sigma N = 1087{,}9$ kW

The motors must deliver $N = 847$ kW and during acceleration, at least 1088 kW.

Take motors: $N = 2 \cdot (0{,}55 \cdot 847)$

$= 2 \cdot 466 = 932$ kW

$n = 783/1000$ rev/min

$S_3 - 100$ percent

$f_a = 180$ percent

Note: To keep the grab well-closed during hoisting you are strongly advised to give the closing motor more 'pull' than the holding motor – 55 percent or 60 percent of the total 'pull'.

Wet sand or soil can produce suction when the grab is drawn out of the material, therefore f_a must be sufficient to counteract this force.

Remark: The acceleration of linear moving masses can also be transferred into the acceleration of rotating masses:

Acceleration of linear moving masses	Transfer
$F_3 = \dfrac{Q \cdot v}{g \cdot t_a}$ kN	$J = \left(\dfrac{30}{\pi} \cdot \dfrac{v}{n}\right)^2 \cdot \dfrac{Q}{\eta}$
$F_3 = \dfrac{660 \cdot 1}{9{,}81 \cdot 2}$	$J = \left(\dfrac{30}{\pi} \cdot \dfrac{1}{783}\right)^2 \cdot \dfrac{66\,000}{0{,}9}$
$= 33{,}6$ kN	$= 10{,}92$ kgm^2
$N_3 = \dfrac{F_3 \cdot v}{\eta}$ kW	$\omega = \dfrac{2 \cdot \pi \cdot n}{60}$ rad/sec

Fig. 3.3.2 4000 ton floating crane 'Asian Hercules'

Acceleration of linear moving masses	Transfer
$N_3 = \dfrac{33{,}6 \cdot 1}{0{,}9}$	$\omega = \dfrac{2 \cdot \pi \cdot 783}{60} = 82\ \mathrm{rad/sec}$
$= 37{,}33\ \mathrm{kW}$	
$M_3 = \dfrac{N_3 \cdot 9550}{n}\ \mathrm{Nm}$	$M_3 = \dfrac{J \cdot \omega}{t_\mathrm{a}}\ \mathrm{Nm}$
$M_3 = \dfrac{33{,}73 \cdot 9550}{783}$	$M_3 = \dfrac{10{,}92 \cdot 82}{2}$
$= 455\ \mathrm{Nm}$	$= 448\ \mathrm{Nm}$

3.4 Calculating the needed power of the trolley travelling motors

In general, we have 3 types of different systems and calculations.

A Direct driven trolleys or motor trolleys.
B Trolleys, which are pulled by wire ropes.

C Rope driven trolleys for grab-unloaders with a main- and an auxiliary trolley.

For the direct driven or motor trolleys account must be taken of the possibility of slip between the direct driven wheel and the rail, under bad weather conditions.

Factors to be considered are:

1. The resistance due to nominal travelling.
2. The resistance due to the current supply- or festoon system.
3. The resistance due to the influence of the wind on the trolley and the load.
4. The resistance due to the acceleration of the rotating masses.
5. The resistance due to the acceleration of the linear moving masses.

For systems A and B, the motor trolleys and rope driven trolleys, we arrive at the following calculation; after checking the main characteristics.

Main characteristics

		Example A	Example B
Trolley travelling speed	m/min m/sec	150 m/min $v = 2{,}5$ m/sec (Full-motor trolley)	210 m/min $v = 3{,}5$ m/sec (Full-rope trolley)
Weight of the trolley	t	$W_1 = 75$ t	$W_1 = 35$ t
Weight of the total load	t	$W_2 = 55$ t	$W_2 = 55$ t
Total weight	t	$Wt = 130$ t	$Wt = 90$ t
Wheel resistance of the trolley wheels	kN/t	$f = 5$ kg/t $= 0{,}05$ kN/t	0,05 kN/t
Efficiency of gearings (and rope sheaves)	η_t		
Full rope trolley or semi-rope trolley of a container crane with the hoisting mechanism on the bridge			
$\eta_t = \eta_{sh} + \eta_{gearings}$			$\eta_t = 0{,}87$ to $0{,}85$

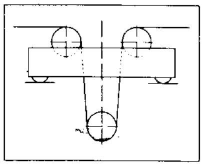


Fig. 3.4.1

		Example A	Example B
Full motor trolley of a crane with the hoisting winch on the trolley			
$\eta_t = \eta_{\text{gearings}}$		$\eta_t = 0{,}90$	
Influence of the wind:			
$F_w = \Sigma(A \cdot c \cdot \eta) \cdot q$	kN		
$q = 150\ \text{N/m}^2$ – $v_w = 15{,}5\ \text{m/sec}$		$F_w = 18\ \text{kN}$	$F_w = 12\ \text{kN}$
$q = 200\ \text{N/m}^2$ – $v_w = 17{,}9\ \text{m/sec}$			
$q = 250\ \text{N/m}^2$ – $v_w = 20\ \text{m/sec}$			
$q = 300\ \text{N/m}^2$ – $v_w = 21{,}9\ \text{m/sec}$			
$q = 400\ \text{N/m}^2$ – $v_w = 25{,}3\ \text{m/sec}$			
Acceleration time:			
$t_a = \ldots.$ sec	sec	$t_a = 6$ sec	$t_a = 4$ sec
Acceleration:			
$a = \ldots.\ \text{m/sec}^2$	m/sec^2	$a = 2{,}5/6 = 0{,}41\ \text{m/sec}^2$	$a = 3{,}5/4 = 0{,}875\ \text{m/sec}^2$
Motor speed n_m	rev/min	$n_m = 1500$ rev/min	
Wheel diameter D	m	$D = 0{,}8$ m	Not important, trolley is rope driven

		Example A	Example B
Reduction between motor and wheel	i		
$i = \frac{n_m \cdot \pi \cdot D}{v}$		$i = \frac{1500 \cdot \pi \cdot 0{,}8}{150}$ $= 25{,}12$	
$J_t = \Sigma_{mom}$ of inertia of the rot. parts	kg m^2		
J_{motors}		×	
$J_{break\ sheaves/couplings}$		×	
$J_{gearbox,\ reduced}$		×	
J_t	kg m^2	$J_t = 6$ kgm^2	
Resistance due to the current supply- or festoon-system: – take 1,5 kN for a rope driven trolley and 3–4 kN for a motor trolley			

3.4.1(A) Direct driven trolleys or motor trolleys; wheel slip control

		kN	kW
1. Resistance due to nominal travelling:			
$F_1 = W_t \cdot f$	kN	$F_1 = 130 \cdot 0{,}05$ $= 6{,}5$ kN	
$N_1 = \frac{F_1 \cdot v}{\eta}$	kW		$N_1 = \frac{6{,}5 \cdot 2{,}5}{0{,}9}$ $= 18{,}05$ kW
2. Resistance due to the festoon system:			
F_2	kN	$F_2 = 3$ kN	
$N_2 = \frac{F_2 \cdot v}{\eta}$	kW		$N_2 = \frac{3 \cdot 2{,}5}{0{,}9}$ $= 8{,}3$ kW

			kN	kW
3. Resistance due to wind:				
	$F_3 = F_w$	kN	$F_3 = 18\,kN$	
	$N_3 = \dfrac{F_3 \cdot v}{\eta}$	kW		$N_3 = \dfrac{18 \cdot 2{,}5}{0{,}9} = 50\,kW$
4. Resistance due to the accelerating of the rotating masses:				
	$\omega = \dfrac{n_m \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{1500 \cdot 2 \cdot \pi}{60} = 157\,rad/sec$	
	$M_4 = \dfrac{J_t \cdot \omega}{t_a}$	Nm	$M_4 = \dfrac{6 \cdot 157}{6} = 157\,Nm$	
	F_4	kN	— (Remains internal in the drive)	
	$N_4 = \dfrac{M_4 \cdot n_m}{9550}$	kW		$N_4 = \dfrac{157 \cdot 1500}{9550} = 24{,}65\,kW$
5. Resistance due to the acceleration of the linear masses:				
	$F_5 = \dfrac{W_t \cdot v}{t_a}$	kN	$F_5 = \dfrac{130 \cdot 2{,}5}{6} = 54{,}16\,kN$	
	$N_5 = \dfrac{F_5 \cdot v}{\eta}$	kW		$N_5 = \dfrac{54{,}16 \cdot 2{,}5}{0{,}9} = 150{,}44\,kW$

Addition: (Motor trolley)	Drive forces on the wheels (kN)	Needed motor power (kW)
1. Nominal travelling	$F_1 = 6{,}5\,\text{kN}$	$N_1 = 18{,}05\,\text{kW}$
2. Festoon system	$F_2 = 3\,\text{kN}$	$N_2 = 8{,}3\,\text{kW}$
3. Wind $q = 150\,\text{N/m}^2$	$F_3 = 18\,\text{kN}$	$N_3 = 50\,\text{kW}$
Total for nominal travelling + wind	$\Sigma F = 27{,}5\,\text{kN}$	$\Sigma N = 76{,}35\,\text{kW}$

Addition: During acceleration	kN	kW
1. Nominal travelling	$F_1 = 6{,}5\,\text{kN}$	$N_1 = 18{,}05\,\text{kW}$
2. Festoon system	$F_2 = 3\,\text{kN}$	$N_2 = 8{,}3\,\text{kW}$
3. Wind $q = 150\,\text{N/m}^2$	$F_3 = 18\,\text{kN}$	$N_3 = 50\,\text{kW}$
4. Acceler. rot. masses, $t_a = 6$ sec	$F_4 = -\text{kN}$	$N_4 = 24{,}65\,\text{kW}$
5. Acceler. linear masses, $t_a = 6$ sec	$F_5 = 54{,}16\,\text{kN}$	$N_5 = 150{,}44\,\text{kW}$
Total, during acceleration:	$\Sigma F_a = 81{,}66\,\text{kN}$ (for control of the slip between rail and wheel)	$\Sigma N_a = 251{,}44\,\text{kW}$

The needed motor power must now be greater than $\Sigma N = 76{,}35\,\text{kW}$ and $\Sigma N_a = 251{,}44/f_a$. f_a is the maximum torque of the motors, which should not be greater than $f_a = 2$ ($M_{max} = 200$ percent of M_{nom}). So:

ΣN must be greater than $\Sigma N = 76{,}35\,\text{kW}$ and ΣN must be greater than

$$\Sigma N_a = \frac{251{,}44}{2} = 125{,}7\,\text{kW}$$

Take $\Sigma N = 126\,\text{kW}$.

If 4 wheels are driven, 4 motors each of $N = 31{,}5\,\text{kW}$ are needed for the motor trolley.

Wheel slip control

The slip between rail and wheels must be controlled for a loaded as well as an unloaded motor driven trolley:

	Loaded trolley	Unloaded trolley
Total wheel-loads	$W_t = 130$ t	$W = 130$ t $- 40$ t (container) $W_t = 90$ t
Nos. of wheels	4	4
Nos. of driven wheels	4	4
Average wheel load	$P = 130/4 = 32{,}5$ t	$P = 90/4 = 22{,}5$ t
Maximum traction force per wheel	$F = 81{,}66/4$ $= 20{,}4$ kN	$F = 81{,}66/4$ $= 20{,}4$ kN
Needed friction coefficient between wheel and rail allowed is $\mu = 0{,}12 = 12$ percent	$\mu = 0{,}1 \cdot F/P$ $= 0{,}1 \cdot \dfrac{20{,}4}{32{,}5}$ $= 0{,}062$ $= 6{,}2$ percent	$M\mu = 0{,}1 \cdot F/P$ $= 0{,}1 \cdot \dfrac{20{,}4}{22{,}5}$ $= 0{,}0906$ $= 9{,}1$ percent

3.4.2(B) Trolleys pulled by wire ropes or rope driven trolleys

(We continue with the trolley with the main characteristics mentioned under Section 3.4.2(A). We repeat these for clarity.)

Trolley travelling speed	m/min	210 m/min
	m/sec	$v = 3{,}5$ m/sec (full rope trolley)
Weight of the trolley	t	$W_1 = 35$ t
Weight of the total load	t	$\underline{W_2 = 55\ \text{t}}$
Total weight	t	$W_t = 90$ t
Efficiency of gearings and rope sheaves		$\eta_t = 0{,}85$

Influence of the wind:		
$F_w = \Sigma(A \cdot c \cdot \eta) \cdot q$	kN	
$q = 150\ \mathrm{N/m^2} - v_w = 15{,}5\ \mathrm{m/sec}$		$F_w = 12\ \mathrm{kN}$
$q = 200\ \mathrm{N/m^2} - v_w = 17{,}9\ \mathrm{m/sec}$		
$q = 250\ \mathrm{N/m^2} - v_w = 20\ \mathrm{m/sec}$		
$q = 300\ \mathrm{N/m^2} - v_w = 21{,}9\ \mathrm{m/sec}$		
$q = 400\ \mathrm{N/m^2} - v_w = 25{,}3\ \mathrm{m/sec}$		
Acceleration time:		
$t_a = \ldots.\ \mathrm{sec}$	sec	$t_a = 4$ sec
Acceleration:		
$a = \ldots.\ \mathrm{m/sec^2}$	m/sec²	$a = 3{,}5/4 = 0{,}875\ \mathrm{m/sec^2}$
$J_t = \Sigma_{mom}$ of inertia of the rot. parts	kg m²	
J_{motors}		×
$J_{brake\ sheaves/coupling}$		×
$J_{gearbox\ reduced}$		×
$J_{total} = J_t$	kg m²	$J_t = 8$ kg m²
Motor	rev/min	$n_m = 1000$ rev/min
Resistance due to festoon system	kN	$F_2 = 1{,}5$ kN

			kN	kW
B – Rope driven trolley: calculation				
1. Resistance due to nominal travelling				
	$F_1 = W_t \cdot f$	kN	$F_1 = 90 \cdot 0{,}06 = 5{,}4\ \mathrm{kN}$	
	$N_1 = \dfrac{F_1 \cdot v}{\eta}$	kW		$N_1 = \dfrac{5{,}4 \cdot 3{,}5}{0{,}85} = 22{,}2\ \mathrm{kW}$
	(For full-supported hoist-wire ropes it can be necessary to calculate approximately twice as much for F_1 and N_1)			

		kN	kW
2. Resistance due to the festoon system:			
F_2	kN	$F_2 = 1{,}5\,\text{kN}$	
$N_2 = \dfrac{F_2 \cdot v}{\eta}$	kW		$N_2 = \dfrac{1{,}5 \cdot 3{,}5}{0{,}85}$ $= 6{,}2\,\text{kW}$
3. Resistance due to wind:			
$F_3 = F_w$	kN	$F_3 = 12\,\text{kN}$	
$N_3 = \dfrac{F_3 \cdot v}{\eta}$	kW		$N_3 = \dfrac{12 \cdot 3{,}5}{0{,}85}$ $= 49{,}4\,\text{kW}$
4. Resistance due to the acceleration of the rotating masses:			
$\omega = \dfrac{n_m \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{1000 \cdot 2 \cdot \pi}{60}$ $= 104{,}66\,\text{rad/sec}$	
$M_a = \dfrac{J_t \cdot \omega}{t_a}$	Nm	$M_a = \dfrac{8 \cdot 104{,}66}{4}$ $M_4 = 209{,}32\,\text{Nm}$	
F_4	kN	— (Remains internal in drive)	
$N_4 = \dfrac{M_a \cdot n_m}{9550}$	kW		$N_4 = \dfrac{209{,}32 \cdot 1000}{9550}$ $= 21{,}92\,\text{kW}$
5. Resistance due to the acceleration of the linear masses			
$F_5 = \dfrac{W_t \cdot v}{t_a}$	kN	$F_5 = \dfrac{90 \cdot 3{,}5}{4}$ $= 78{,}75\,\text{kN}$	
$N_5 = \dfrac{F_5 \cdot v}{\eta}$	kW		$N_5 = \dfrac{78{,}75 \cdot 3{,}5}{0{,}85}$ $= 324{,}4\,\text{kW}$

Addition: (rope trolley)	Drive forces in the wire ropes (kN)	Needed motor power (kW)
1. Nominal travelling	$F_1 = 5{,}4\,\mathrm{kN}$	$N_1 = 22{,}2\,\mathrm{kW}$
2. Festoon system	$F_2 = 1{,}5\,\mathrm{kN}$	$N_2 = 6{,}2\,\mathrm{kW}$
3. Wind $q = 150\,\mathrm{N/m^2}$	$F_3 = 12\,\mathrm{kN}$	$N_3 = 49{,}4\,\mathrm{kW}$
Total for nominal travelling + wind	$\Sigma F = 18{,}9\,\mathrm{kN}$	$\Sigma N = 77{,}8\,\mathrm{kW}$

During acceleration	Drive forces in the wire ropes (kN)	Needed motor power (kW)
1. Nominal travelling	$F_1 = 5{,}4\,\mathrm{kN}$	$N_1 = 22{,}2\,\mathrm{kW}$
2. Festoon system	$F_2 = 1{,}5\,\mathrm{kN}$	$N_2 = 6{,}2\,\mathrm{kW}$
3. Wind $q = 150\,\mathrm{N/m^2}$	$F_3 = 12\,\mathrm{kN}$	$N_3 = 49{,}4\,\mathrm{kW}$
4. Acceler. rot. masses, $t_a = 4$ sec	$F_4 = -\mathrm{kN}$	$N_4 = 21{,}92\,\mathrm{kW}$
5. Acceler. linear masses, $t_a = 4$ sec	$F_5 = 78{,}75\,\mathrm{kN}$	$N_5 = 324{,}3\,\mathrm{kW}$
Total during acceleration	$\Sigma F = 97{,}65\,\mathrm{kN}$	$\Sigma N = 424\,\mathrm{kW}$

The needed motor power must be greater than

$$\Sigma N = 77{,}8\,\mathrm{kW} \quad \text{and} \quad \Sigma N = 424/f_a.$$

f_a is again the maximum torque factor of the motors, which should – in this case – not be greater than $f_a = 2{,}2$. ($M_{max} = 220$ percent of M_{nom}.) So:

ΣN must be greater than $\Sigma N = 77{,}8$ kW and ΣN must be greater than

$$\Sigma N = \frac{424}{2{,}2} = 192{,}7\,\mathrm{kW}$$

The slip control between rails and wheels need not be done.

3.4.3(C) Rope driven trolleys for grab unloaders with a main and auxiliary trolley

This type of high throughput unloader always has high-speed rope trolleys with acceleration and deceleration times of only approximately 3 sec.

Therefore it is necessary to take the extra rope-pull of the accelerating and swinging load into account when calculating the requisite motor power. (See Fig. 2.1.2 for a diagram of this assembly.)

Factors to be considered are:

1. The resistance due to nominal travelling of the main-trolley plus loaded grab.
2. The resistance due to nominal travelling of the auxiliary trolley (at half the speed of the main trolley).
3. The resistance due to the influence of the wind on the trolley and the load.
4. The resistance due to the acceleration of the rotating parts.
5. The resistance due to the acceleration of the main trolley.
6. The resistance due to the acceleration of the auxiliary trolley.
7. The resistance through the sharp acceleration of the loaded grab which swings to its extremes.

Main characteristics: example

Grab travelling speed	m/min m/sec	260 m/min $v = 4{,}33$ m/sec
Weight of the main trolley	mt	$W_1 = 16$ t
Weight of the loaded grab	mt	$W_2 = 45$ t
Weight of the auxiliary trolley	mt	$W_3 = 13$ t
Wheel resistance of the trolley wheels	kN/t	$f = 5$ kg/t $= 0{,}05$ kN/t
Efficiency of all gearings and rope sheaves	η_t	$\eta_t = 0{,}85$
Influence of the wind		
$F_w = \Sigma(A \cdot C \cdot \eta_2) \cdot q$	kN	
$q = 150$ N/m^2 $- v_w = 15{,}5$ m/sec		
$q = 200$ N/m^2 $- v_w = 17{,}9$ m/sec		
$q = 250$ N/m^2 $- v_w = 20$ m/sec		$F_w = 6{,}3$ kN
$q = 300$ N/m^2 $- v_w = 21{,}9$ m/sec		
$q = 400$ N/m^2 $- v_w = 25{,}3$ m/sec		

Acceleration time: $t_a = \ldots .$ sec	sec	$t_a = 3$ sec
Acceleration: $a = \ldots .$ m/sec^2	m/sec^2	$a = 1{,}444$ m/sec^2
Motor speed	rev/min	$n_m = 1500$ rev/min
$J_t = \Sigma_{mom}$ of inertia of the rot. parts	kg m^2	
J_{motors}		$J_m = 9$ kg m^2
$J_{brakesheaves/couplings}$		$J_b = 8$ kg m^2
$J_{gearbox,\ reduced}$		$J_{gb} = 2$ kg m^2
$J_{total} = J_t$	kg m^2	$J_t = 19$ kg m^2

Calculation:		Drive forces in the wire ropes (kN)	Needed motor power (kW)
1. Resistance due to nominal travelling of the main trolley plus loaded grab:			
$F_1 = (W_1 + W_2) \cdot f$	kN	$F_1 = (16 + 45) \cdot 0{,}05 = 3{,}05$ kN	
$N_1 = \dfrac{F_1 \cdot v}{\eta}$	kW		$N_1 = \dfrac{3{,}05 \cdot 4{,}33}{0{,}85} = 15{,}53$ kW
2. Resistance due to nominal travelling of the auxiliary trolley:			
$F_2 = W_3 \cdot f$	kN	$F_2 = 13 \cdot 0{,}05 = 0{,}65$ kN	
$N_2 = \dfrac{F_2 \cdot 0{,}5 \cdot v}{\eta}$	kW		$N_2 = \dfrac{0{,}65 \cdot 0{,}5 \cdot 4{,}33}{0{,}85} = 1{,}65$ kW

Calculation:		Drive forces in the wire ropes (kN)	Needed motor power (kW)
3. Resistance due to the wind:			
$F_3 = F_w$	kN	$F_3 = 6{,}3\ \text{kN}$	
$N_3 = \dfrac{F_3 \cdot v}{\eta}$	kW		$N_3 = \dfrac{6{,}3 \cdot 4{,}33}{0{,}85}$ $= 32{,}1\ \text{kW}$
4. Resistance due to the acceleration of the rotating masses:			
$\omega = \dfrac{n_m \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{1500 \cdot 2 \cdot \pi}{60}$ $= 157\ \text{rad/sec}$	
$M_a = \dfrac{J_t \cdot \omega}{t_a}$	Nm	$M_a = \dfrac{19 \cdot 157}{3}$ $M_a = 994\ \text{Nm}$	
F_4	kN	— (remains internal in the drive)	
$N_4 = \dfrac{M_a \cdot n_m}{9550}$	kW		$N_4 = \dfrac{994 \cdot 1500}{9550}$ $= 156\ \text{kW}$
5. Resistance due to the acceleration of the main trolley:			
$F_5 = \dfrac{W_1 \cdot v}{t_a}$	kN	$F_5 = \dfrac{16 \cdot 4{,}33}{3}$ $= 23{,}1\ \text{kN}$	
$N_5 = \dfrac{F_5 \cdot v}{\eta}$	kW		$N_5 = \dfrac{23{,}1 \cdot 4{,}33}{0{,}85}$ $= 117{,}7\ \text{kW}$

Calculation		Drive forces in the wire ropes (kN)	Needed motor power (kW)
6. Resistance due to the acceleration of the auxiliary trolley:			
$F_6 = \dfrac{W_3 \cdot 0{,}5 \cdot v}{t_a}$	kN	$F_6 = \dfrac{13 \cdot 0{,}5 \cdot 4{,}33}{3} = 9{,}4\,\text{kN}$	
$N_6 = \dfrac{F_6 \cdot 0{,}5 \cdot v}{\eta}$			$N_6 = \dfrac{9{,}4 \cdot 0{,}5 \cdot 4{,}33}{0{,}85} = 23{,}94\,\text{kW}$
7. Resistance due to the sharp acceleration of the loaded grab: Take $\alpha_1 = 8{,}3$ degrees; $2 \cdot \alpha_1 = 16{,}6$ degrees			
$F_7 = W_2 \cdot 10 \cdot \sin(2\alpha)$	kN	$F_7 = 45 \cdot 10 \cdot 0{,}285 = 128{,}25\,\text{kN}$	
$N_7 = \dfrac{F_7 \cdot v}{\eta}$	kW		$N_7 = \dfrac{128{,}25 \cdot 4{,}33}{0{,}85} = 653{,}3\,\text{kW}$

$(F_a = m \cdot a = \dfrac{450}{10} \cdot \dfrac{4{,}33}{3} = 64{,}95\,\text{kN}$

$\sin \alpha_1 = 64{,}95/450 = 0{,}1443$

$\alpha_1 = 8{,}3$ degrees

Take $\alpha = 2 \times 8{,}3$ degrees $= 16{,}6$ degrees in order to calculate the swing of the grab.
Taking this great swing into account is necessary.
The motor must be able to develop a great torque!)

Addition: (bulk unloader)	Drive forces in the wire ropes (kN)	Needed motor power (kW)
1. Nom. trav. of main trolley plus grab	$F_1 = 3{,}05\,\text{kN}$	$N_1 = 15{,}53\,\text{kW}$
2. Nom. trav. aux. trolley	$F_2 = 0{,}65\,\text{kN}$	$N_2 = 1{,}65\,\text{kW}$
3. Wind, $q = 250\,\text{N/m}^2$	$F_3 = 6{,}3\,\text{kN}$	$N_3 = 32{,}1\,\text{kW}$
4. Acceler. rot. masses, $t_a = 3$ sec	$F_4 =$ –	$N_4 = 156\,\text{kW}$
5. Acceler. main trolley	$F_5 = 23{,}1\,\text{kN}$	$N_5 = 117{,}7\,\text{kW}$
6. Acceler. aux. trolley	$F_6 = 9{,}4\,\text{kN}$	$N_6 = 23{,}94\,\text{kW}$
7. Acceler. of grab ($2\alpha_1 = 16{,}6$ degrees)	$F_7 = 128{,}25\,\text{kN}$	$N_7 = 653{,}3\,\text{kW}$
8. Total during acceleration	$\Sigma F = 170{,}75\,\text{kN}$	$\Sigma N = 1000{,}2\,\text{kW}$

If we take $f_a = 2{,}5$ as the maximum torque factor, the motor power should be:

$$N_m = \frac{1000{,}2}{2{,}5} = 400\,\text{kW}$$

$$f_a = 2{,}5;\; n_m = 1500\,\text{rev/min}$$

Due to the very short cycle time of approximately 45 sec, the rating of the motor should be approximately 80 percent to 100 percent.

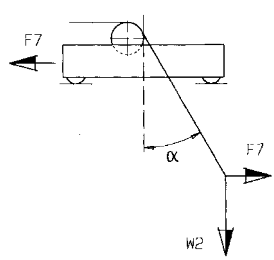


Fig. 3.4.2

Fig. 3.4.3 The swinging grab of an unloader

Fig. 3.4.4 Grab of a floating unloader

3.5 Hoisting the boom; calculating the power needed for the boom hoist motor

Make a schematic diagram (as Fig. 3.5.1) and estimate the weight of the boom and it's centre of gravity.

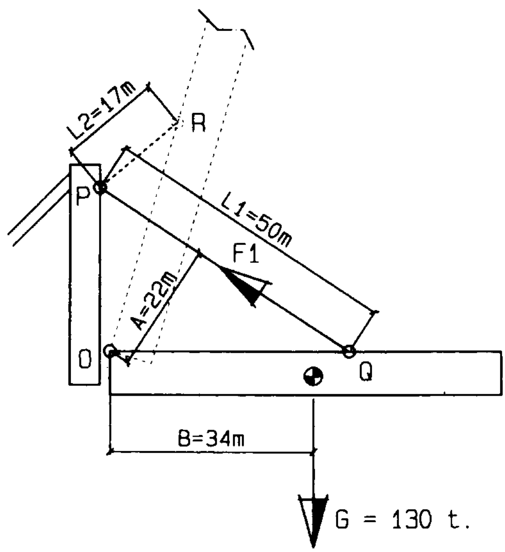


Fig. 3.5.1

For an example we use the following:

– Centre of rotation = the hinge point of the boom = O	
– Weight of the boom:	$G = 130$ t
– Distance from O to the centre of gravity:	$B = 34$ m
– (With boom down):	$L_1 = 50$ m
– (With boom up):	$L_2 = 17$ m
– Distance from hinge point to centre of the line PQ:	$A = 22$ m
– The asked for time for boom-hoisting:	$t_1 = 5{,}5$ min
– Creeping time for the boom when starting and ending the hoist movement, including latching:	$t_2 = 1$ min
– Time for really hoisting the boom:	$t_3 = t_1 - t_2 = 5{,}5 - 1 = 4{,}5$ min

– Average speed: $v = \dfrac{PQ - PR}{t_3} = \dfrac{L_1 - L_2}{t_3}$

$$v = \frac{50 - 17}{4{,}5} = 7{,}33 \text{ m/min}$$

– Total efficiency of the wirerope sheaves, the drum and the gearbox: $\eta = 0{,}86$

– Force in the boom-hoist-tackle, when starting to hoist the boom: $F_1 = \dfrac{G \cdot B}{A}$

$$F_1 = \frac{130 \cdot 34}{22}$$

$$F_1 = 201 \text{ t} = 201\,000 \text{ kg}$$

– Required motor power: $N = \dfrac{F_1 \cdot v}{60 \cdot 75 \cdot 1{,}36 \cdot \eta} \text{ kW}$

$$N = \frac{201\,000 \cdot 7{,}33}{60 \cdot 102 \cdot 0{,}86}$$

$$N = 280 \text{ kW}$$

(During the hoisting of the boom the force F_1 becomes lower as the distance A becomes greater.)
(The influence of the wind from the rear becomes greater when the boom is hoisted.)
Take the motor power: $N = 280$ kW
$n = 1500$ rev/min
25 percent; eventual 40 percent rating
$f_a = 160$ percent.
The tackle of the boom hoist mechanism is 2 times Z. $2 \cdot Z = 2 \cdot 9 = 18$. This means that the force F_1 is taken by a bundle of $2 \cdot 9 = 18$ wire ropes.
The efficiency of the sheaves before the drum is: $\eta_s = 0{,}91$.
The force in each of the 2 wire ropes, which are connected to the boom hoist drum is then:

$$F_2 = \frac{F_1}{2 \cdot Z \cdot \eta_s}$$

$$F_2 = \frac{201\,000}{2 \cdot 9 \cdot 0{,}91} = 12\,271 \text{ kg}$$

Note: When a computer calculation of the wire rope forces during boom-hoisting is made, and the weight of the underpart of the boom-forestays is taken into account, it will be found that these underparts give some assistance in lifting the first few degrees from the horizontal. This means that the force in the boom-hoist tackle is reduced over the first 6 degrees by some 7 percent, thus using less motor power. This phenomenon can help in difficult cases.

Normally the wire ropes of the boom-hoist mechanisms will offer a safety factor of about six, against rupture. However, a boom-hoist mechanism is a system that can give rise to difficulties if one of the two wire ropes should fail. Cases of a boom or even a whole crane collapsing when a wire rope breakdown occurs are well known in the annals of crane history. If one wire rope should fail, the safety factor against rupture of the remaining wire rope would only be three which is not high enough for safe operation.

The safest manner for the reeving-in of the boom-hoist mechanism is shown in Fig. 3.5.2.

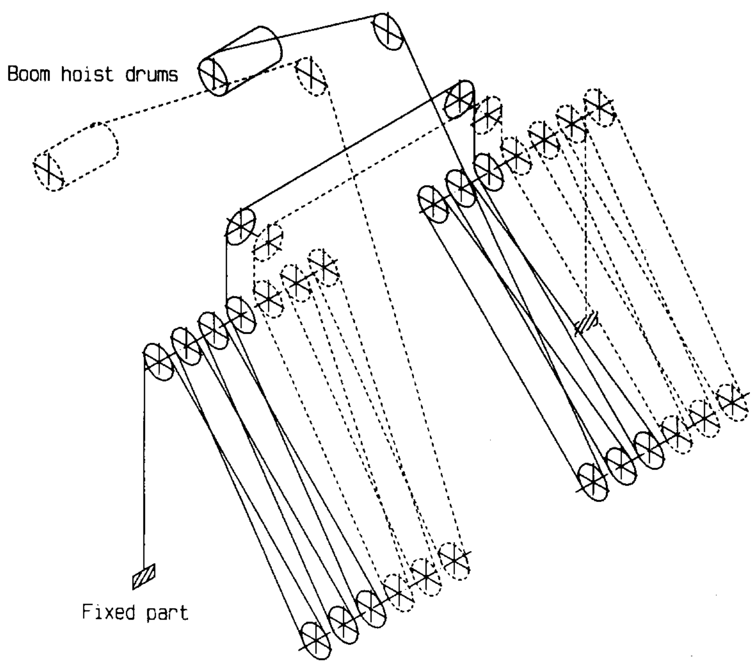


Fig. 3.5.2 Ideal reeving system for boom-hoist mechanism

3.6 Calculating the needed power of the crane-travelling motors. Wheelslip control – how to calculate the forces through skewing of the crane and trolley

Factors to be considered are:

1. The resistance due to nominal travelling.
2. The resistance due to the influence of the wind.
3. The resistance due to the acceleration of the rotating masses.
4. The resistance due to the acceleration of the linear moving masses.

Main characteristics

		Example Loaded Crane
Weight of crane	mt	$W_1 = 1300$ t
Weight of load	mt	$W_2 = 40$ t
Total weight	mt	$W_t = 1340$ t
Crane travelling speed	m/min	45 m/min
	m/sec	$v = 0{,}75$ m/sec
Total efficiency of the gearings	η_t	$\eta_t = 0{,}9$
Wheel resistance of the crane travelling wheels	kN/t	$f = 5$ kg/t $= 0{,}05$ kN/t
Influence of the (side) wind:		
$F_w = \Sigma(A \cdot C \cdot \eta_2) \cdot q$	kN	
$q = 150$ N/m^2 $- v_w = 15{,}5$ m/sec		
$q = 200$ N/m^2 $- v_w = 17{,}9$ m/sec		
$q = 275$ N/m^2 $- v_w = 21$ m/sec		$F_w = 510$ kN
$q = 400$ N/m^2 $- v_w = 25{,}3$ m/sec		
Acceleration time:		
$t_a = \ldots.$ sec	sec	$t_a = 6$ sec

		Example Loaded Crane
Acceleration		
$a = \ldots.\ \mathrm{m/sec^2}$	$\mathrm{m/sec^2}$	$a = \frac{v}{t} = \frac{0{,}75}{6}$ $= 0{,}125\ \mathrm{m/sec^2}$
Motor speed: n_{m}	rev/min	$n_{\mathrm{m}} = 1800$ rev/min
Wheel diameter: D_{w}	m	$D_{\mathrm{w}} = 0{,}9$ m
Reduction between motor and wheel:		
$i = \frac{n_{\mathrm{m}} \cdot \pi \cdot D_{\mathrm{w}}}{v}$		$i = \frac{1800 \cdot \pi \cdot 0{,}9}{45} = 113$
$J_{\mathrm{t}} = \Sigma_{\mathrm{mom}}$ of inertia of the rot. parts	$\mathrm{kg\,m^2}$	
J_{motors}		×
$J_{\text{brake sheaves/couplings}}$		×
$J_{\text{gearbox, reduced}}$		×
$J_{\mathrm{total}} = J_{\mathrm{t}}$		$J_{\mathrm{t}} = 2\ \mathrm{kg\,m^2}$

Calculation		kN	kW
1. Resistance due to nominal travelling:			
$F_1 = W_{\mathrm{t}} \cdot f$	kN	$F_1 = 1340 \cdot 0{,}05$ $= 67$ kN	
$N_1 = \frac{F_1 \cdot v}{\eta_{\mathrm{t}}}$	kW		$N_1 = \frac{67 \cdot 0{,}75}{0{,}9}$ $= 55{,}8$ kW
2. Resistance due to wind:			
$F_2 = F_{\mathrm{w}}$	kN	$F_2 = 510$ kN	
$N_2 = \frac{F_2 \cdot v}{\eta_{\mathrm{t}}}$	kW		$N_2 = \frac{510 \cdot 0{,}75}{0{,}9}$ $= 425$ kW

Calculation		kN	kW
3. Resistance due to the acceleration of the rotating masses:			
$\omega = \dfrac{n_m \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{1800 \cdot 2 \cdot \pi}{60}$ $= 188{,}4$ rad/sec	
$M_a = \dfrac{J_t \cdot \omega}{t_a}$	Nm	$M_a = \dfrac{2 \cdot 188{,}4}{6}$ $= 62{,}8$ Nm	
F_3 (remains internal in the drive)	kN	—	
$N_3 = \dfrac{M_a \cdot n_m}{9550}$	kW		$N_3 = \dfrac{62{,}8 \cdot 1800}{9550}$ $= 11{,}8$ kW
4. Resistance due to the acceleration of the linear moving masses:			
$F_4 = \dfrac{W_t \cdot v}{t_a}$	kN	$F_4 = \dfrac{1340 \cdot 0{,}75}{6}$ $= 167{,}5$ kN	
$N_4 = \dfrac{F_4 \cdot v}{\eta_t}$	kW		$N_4 = \dfrac{167{,}5 \cdot 0{,}75}{0{,}9}$ $= 139{,}5$ kW

Addition	Lateral forces on the travelling wheels	Motor power
1. Nominal travelling	$F_1 = 67$ kN	$N_1 = 55{,}8$ kW
2. Wind $q = 275$ N/m^2	$F_2 = 510$ kN	$N_2 = 425$ kW
Total, for nominal travelling + wind	$\Sigma F = 577$ kN	$\Sigma N = 480{,}8$ kW

Addition	Lateral forces on the travelling wheels	Motor power
During acceleration:		
1. Nominal travelling	$F_1 = 67\,\text{kN}$	$N_1 = 55{,}8\,\text{kW}$
2. Wind, $q = 275\,\text{N/m}^2$	$F_2 = 510\,\text{kN}$	$N_2 = 425\,\text{kW}$
3. Acceler. rot. masses, $t_a = 6$ sec	$F_3 = -\text{kN}$	$N_3 = 11{,}8\,\text{kW}$
4. Acceler. lin. masses, $t_a = 6$ sec	$F_4 = 167{,}5\,\text{kN}$	$N_4 = 139{,}5\,\text{kW}$
Total during acceleration	$\Sigma F_a = 744{,}5\,\text{kN}$	$\Sigma N_4 = 632{,}1\,\text{kW}$
Overload factor of motors during acceleration		
$f_a = M_{max} : M_{nom}$	$f_a = 160$ percent	$\Sigma N = \frac{632{,}1}{1{,}6} = 395\,\text{kW}$

Take motors, each 20 kW; so 24 motors.
Total available power $N = 24 \cdot 20 = 480\,\text{kW}$.

The influence of a slope

If a crane has to drive up a slope, an additional resistance has to be overcome. Assume that a rubber tyred gantry (RTG) has to run against the slope of $\alpha = 1$ degree. The total weight of the loaded RTG is $Q_t =$ 165 tons; the crane travelling speed is

$$v = 135\,\text{m/min} = 2{,}25\,\text{m/sec}$$

$$F_{slope} = Q_t \cdot 10^3 \cdot g \cdot \sin\alpha (N)$$

$$F_{slope} = 165 \cdot 10^3 \cdot 9{,}81 \cdot 0{,}0174 = 28\,164\,N = 28{,}164\,\text{kN}$$

$$N_{slope} = \frac{F_{slope} \cdot v}{\eta} = \frac{28{,}164 \cdot 2{,}25}{0{,}9} = 70{,}4\,\text{kW}$$

Wheelslip control

- Check the minimum wheel load on a driven crane wheel: $P_1 =$ . . . . kN
- Check the maximum driving force that the crane travelling motor can exert on the circumference of the driven wheel (traction force):

$$P_2 = \frac{f_a \cdot N \cdot \eta}{v}\,\text{kN}$$

– Friction coefficient between rail and wheel: $mu = P_2/P_1$
– Allowed is $mu = 0{,}12$.

Skewing of the crane and trolley

Cranes, and trolleys, can skew. This can cause severe wear and tear of the rails and the travelling wheels. The FEM standards mention the following (in booklet 2) about skew:

2.2.3.3 Transverse reactions due to rolling action

When two wheels (or two bogies) roll along a rail, the couple formed by the horizontal forces normal to the rail shall be taken into consideration. The components of this couple are obtained by multiplying the vertical load exerted on the wheels (or bogies) by a coefficient λ which depends upon the ratio of the span ρ to the wheelbase a.[1]

[1] By 'wheelbase' is understood the centre distance between the outermost pairs of wheels, or, in the case of bogies, the centre distance between the fulcrum pins on the crane structure of the two bogies or bogie systems. Where horizontal guiding wheels are provided, the wheelbase shall be the distance between the rail contact points of two horizontal wheels.

As shown in the graph [Fig. 3.6.2], this coefficient lies between 0,05 and 0,2 for ratios of ρ/a between 2 and 8.

However DIN and other standards give a more complex calculation about the horizontal forces through skewing. This calculation leads to greater forces than those mentioned in Fig. 3.6.2.

Advice

– Take the skew forces on crane- and trolley wheels as a minimum as 10 percent of the maximum wheel load.
– Take the skew forces on crane- and trolley wheels as 20 percent of the maximum wheel load for cranes and trolleys running $v =$ 150 m/m or more.
– Also check the calculations according to Fig. 3.6.2.

In order to keep the skewing forces on a crane travelling mechanism under reasonable control, the length/width ratio, being the relation between the railspan or railgauge, and the centre distance between the fulcrum pins of the crane travelling mechanism under each corner ρ:a or L:B, should be at least 6:1.

Fig. 3.6.1

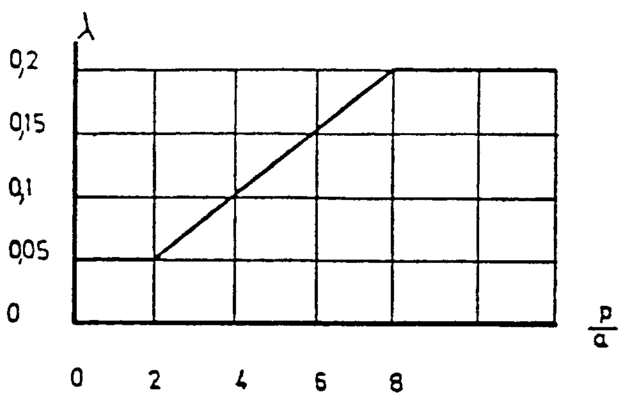


Fig. 3.6.2

3.7 The rating of the motors

Up to now, we have mentioned the following for the motors:

- the power in kW;
- the torque in Nm;
- the number of revolutions per minute;
- the starting torque, f_a, being the torque which the motor and the drive can develop during a certain number of seconds, when accelerating the crane or trolley.

For a crane or trolley motor the normal torque–speed diagram of a DC-Full Thyristor motor or an AC-Frequency controlled motor is:

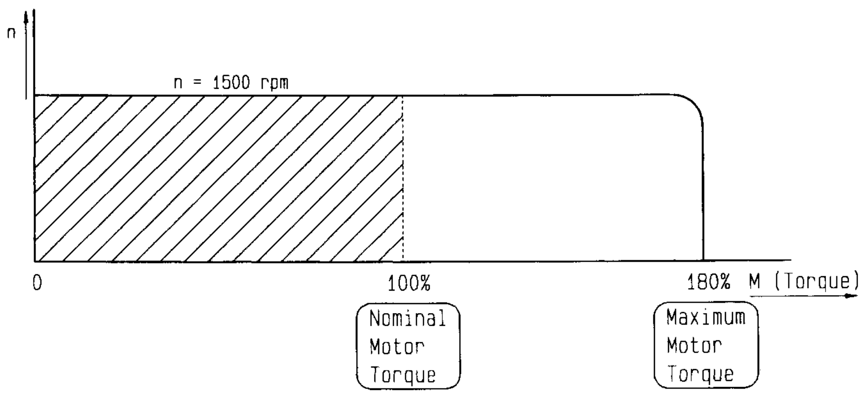


Fig. 3.7.1 Torque–speed diagram

$$M = \frac{N \cdot 9550}{n} \text{Nm}$$

where

M = motor torque in Nm;
N = the number of kW;
n = the number of revolutions per minute of the motor.

The motor of a crane or trolley runs intermittently.

If we consider the trolley travel motor of a bulk unloader, the cycle diagram in Fig. 3.7.2 is produced.

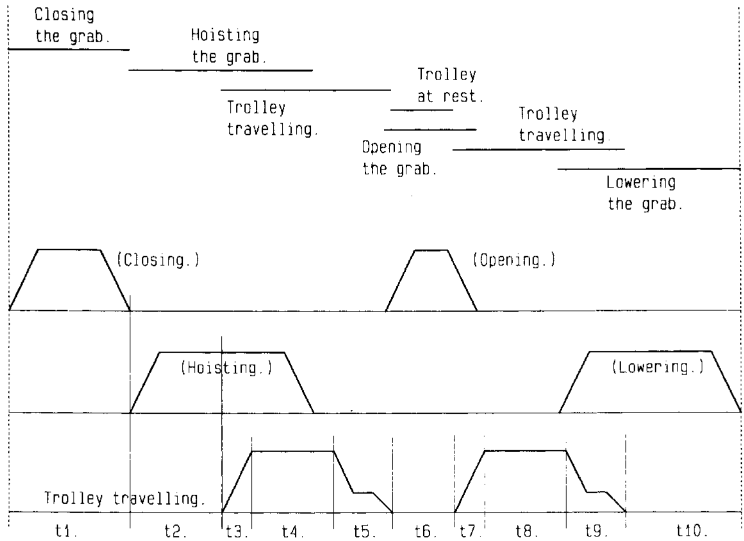


Fig. 3.7.2 Cycle diagram

Cycle

Only for the trolley travel motor

1. Grab digs in and closes — t_1 sec — $M_1 = 0$
2. Grab is hoisted to above the coaming — t_2 sec — $M_2 = 0$
3. Grab is hoisted. Accelerating the trolley with full grab — t_3 sec — $M_3 = f_a \cdot M$
4. Grab is hoisted to max. level and hoist movement stops. Travelling of the trolley with full grab at nom. speed — t_4 sec — $M_4 = M$
5. Grab is opened above the hopper. Decelerating the trolley with full grap to zero speed — t_5 sec — $M_5 = f_a \cdot M$
6. Grab is further opened. Trolley does not move — t_6 sec — $M_6 = 0$
7. Grab remains open. Trolley, with empty grab, accelerates toward the vessel — t_7 sec — $M_7 = f_a \cdot M$

8.	Travelling of the trolley at nominal speed. At certain moment the grab is lowered	t_8 sec	$M_8 = \sim 0{,}8 \cdot M$
9.	Decelerating the trolley with empty grab to zero speed. Grab is further lowered into the vessel	t_9 sec	$M_9 = f_a \cdot M$
10.	Trolley is at rest. Grab is lowered into the hatch and into the material	t_{10} sec	$M_{10} = 0$

We can deduce from this scheme the rating R:

$$R = \frac{\cancel{t_1} + \cancel{t_2} + t_3 + t_4 + t_5 + \cancel{t_6} + t_7 + t_8 + t_9 + \cancel{t_{10}}}{\Sigma(t_1 \div t_{10})}$$

for t_1; t_2; t_6 and t_{10} $\quad M = 0$.

The normal motor ratings are:

	Container quay cranes %	Container stacking cranes %	Grab-unloaders %
Hoisting	60	60*	89–90
Trolley travelling	60	40–60	80–90
Trolley slewing	40	40	–
Boom hoisting	25–40	–	25–40
Crane travelling	40	60	60

	Level luffing grabbing crane %	Level luffing general cargo crane %
Hoisting	60	40–60
Luffing	60	40–60
Slewing	60	40–60
Crane travelling	25–40	25–40

3.8 The root-mean-square calculation

FEM, booklet 5 of 1998 mentions:

5.8.1.3 THERMAL CALCULATION OF THE MOTOR

5.8.1.3.1 Mean equivalent torque

In order to carry out the thermal calculation, the mean equivalent torque must be determined as a function of the required torque during the working cycles, by the formula:

$$M_{med} = \sqrt{\frac{M_1^2 \cdot t_1 + M_2^2 \cdot t_2 + M_3^2 \cdot t_3 + \cdots + M_8^2 \cdot t_8 + M_9^2 \cdot t_9 + M_{10}^2 \cdot t_{10}}{\Sigma t_1 \div t_{10}}}$$

t_1, t_2, t_3, ... t_n are the periods during which the different torque values are produced; periods of rest are not taken into account.

M_1, M_2, M_3, ... M_n are the calculated torque values, in taking into account all the inertia forces including the one of the rotor mass of the motor.

In the case of variable loads, at least a maximum of 10 successive working cycles for the predimensioning, must be taken into account (see definition 2.1.2.2.).

However this RMS system should never be used to try to reduce the motor power to a level lower than that calculated by the earlier methods. (This RMS calculation is more or less obsolete and should only be used for DC systems.)

3.9 The current supply of a crane by a diesel generator set: calculation methods and warnings

Many cranes have a diesel generator set, mounted on the crane itself. In this case, the current supply by a high or medium voltage net is avoided and the crane is made totally independent.

The diesel generator set itself does not provide difficult problems, however some issues have to be addressed before the most suitable generator set can be installed. The diesel builders commonly use the following notations:

– *Stand-by power rating*: This is the ‘top’ power which the diesel can deliver. It can be delivered over a short period of several hours for a restricted number of times per year.
– *Continuous rating*: This is the power which the diesel can deliver continuously for instance for driving a ship. The ‘continuous rating’ is approximately 70–80 percent of the ‘stand-by rating’.

– *Prime power rating*: This must be defined as 'the net prime power at the flywheel; no fan losses'. It is then the mechanical power that the diesel is delivering on its flywheel. The diesel can for instance deliver this power to a generator which is driving a crane. This generator has to deliver power to the motors of the mechanisms in the crane. The kW loads which the different mechanisms need vary during each cycle of the crane. The 'Prime rating' of the diesel should cover these needs.
– *Load in one step*: This is the load which the diesel can take if a sudden load is asked from the generator set. It is approximately 60 percent of the stand-by rating; it then gives a dip of the frequency of the generator of approximately 10 percent. If the diesel is turbo-loaded, it takes approximately 2 seconds for the turbos to give the electronic controlled diesel the necessary extra power to recover, unless a dummy load of resistances gives a particular bottom load to the diesel, which forces the turbo to run continuously.

Summarizing (example):

1.	Stand-by power rating ('*x*' hours, '*y*' times per year)	900 kW
2.	Continous rating (e.g. driving a ship)	$0{,}7 \cdot 900 = 630$ kW up to $0{,}8 \cdot 900 = 720$ kW
3.	Prime power rating ('net prime power at the flywheel; no fan losses')	$0{,}9 \cdot 900 = 810$ kW
4.	Load in one step rating	$0{,}6 \cdot 900 = 540$ kW

These figures are given for example only; they are dependent on the characteristics which the diesel manufacturers provide. Let us assume that a container quay crane has to be driven by a diesel generator set which is mounted on the crane itself.

To be driven are:

– the hoisting mechanism;
– the trolley travelling mechanism;
– the crane travelling mechanism;
– (incidentally) the boom-hoist mechanism;
– all auxiliary mechanisms, such as lighting, heating, air conditioning and elevator or lift.

Note: The carbon deposit in the cylinders and the turbocharger as well as the lube-oil slurry pollute the diesel and decrease its output.

The scheme is shown in Fig. 3.9.1.

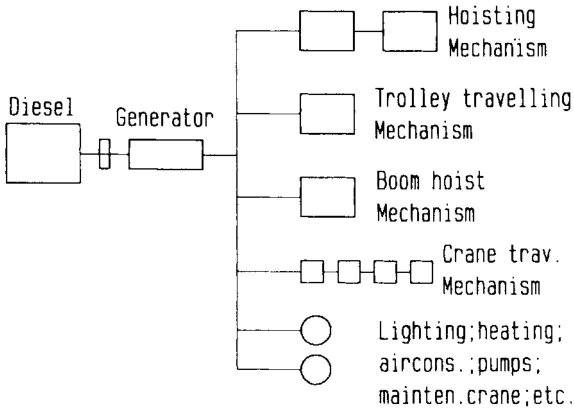


Fig. 3.9.1 Scheme of a diesel generator set

(a) The Direct Current Full Thyristor system

With a Direct Current Full Thyristor system (DC-FT) the normal torque–speed diagram for the electrical motor of the mechanism is as follows.

The current that the generator behind the diesel has to deliver is proportional to the required power that the electrical motor of the mechanism has to give (the cos-phi during acceleration varies between 0 and 0,75).

The DC-FT motor asks approximately 160 percent current when the motorspeed $n = 0$ and $M = 160$ percent.

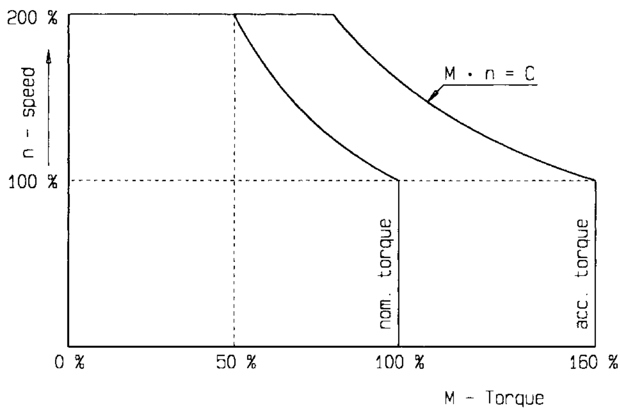


Fig. 3.9.2 DC FT torque–speed diagram

The necessary power in kW is the product of $n \cdot M$; in this case, 1,6 is the factor indicating the maximum torque during acceleration.

The possibilities of simultaneous working mechanisms are as Table 3.9.2. A diesel with an output of approximately 1450 kW Prime Power Rating will do the job for this DC-FT installation, when the f_a factor is in accordance with the figures mentioned in Table 3.9.2. Bear in mind the extra loss of power of a diesel in high temperatures or difficult climatic conditions.

Feeding back into a diesel should be considered. Fortunately, the number of motor kWs to be absorbed by the diesel, can then be multiplied with the efficiencies. The diesel itself can take 10 to 14 percent of the rated output as regenerative loading, depending on the internal friction of the engine.

Table 3.9.1

DC-FT	*A*	*B*	*C*	*D*	*E*
	Hoisting mechanism	*Trolley travelling mechanism*	*Boom hoist mechanism*	*Crane travel mechanism*	*Auxiliaries (lighting, heating, air conditioning, maintenance crane; pumps)*
Number of kiloWatts	600 kW $f_a = 1{,}6$	200 kW $f_a = 2$	250 kW $f_a = 2$	400 kW $f_a = 2$	80 kW
Efficiency of the motor(s) etc.	0,9	0,85	0,85	0,85	0,9
Efficiency between motor(s) and generator	0,95	0,95	0,95	0,9	1
Efficiency of the generator	0,95	0,95	0,95	0,95	0,95
Efficiency between generator and diesel	1 (or 0,9 if there is a gearbox between generator and diesel)				
	η_1	η_2	η_2	η_3	η_4

The total efficiency is:

$\eta_1 = 0{,}9 \cdot 0{,}95 \cdot 0{,}95 \cdot 1 = 0{,}812$

$\eta_2 = 0{,}85 \cdot 0{,}95 \cdot 0{,}95 \cdot 1 = 0{,}767$

$\eta_3 = 0{,}85 \cdot 0{,}9 \cdot 0{,}95 \cdot 1 = 0{,}726$

$\eta_4 = 0{,}9 \cdot 1 \cdot 0{,}95 \cdot 1 = 0{,}855$

Table 3.9.2

	DC-FT	kW	*Necessary diesel output* (kW)
I	A. Hoisting/accelerating	600 · 1,6 = 960	960 : 0,812 = 1182
	B. Auxiliaries	= 80	80 : 0,855 = 93,5
		Total = 1040	Diesel output = 1275,5
II	A. Hoisting	= 600	600 : 0,812 = 739
	B. Trolley travelling/accelerating	200 · 2 = 400	400 : 0,767 = 521.5
	E. Auxiliaries	= 80	80 : 0,855 = 93,5
		Total = 1080	Diesel output = 1354
III	B. Trolley travelling	= 200	200 : 0,767 = 261
	D. Crane travelling/accelerating	400 · 2 = 800	800 : 0,726 = 1102
	E. Auxiliaries	= 80	80 : 0,855 = 93,5
		Total = 1080	Diesel output = 1456,5
IV	D. Crane travelling (against storm)	400 · 2 = 800	800 : 0,726 = 1102
	E. Auxiliaries	= 80	80 : 0,855 = 93,5
		Total = 880	Diesel output = 1195,5

In our case we consider the following for feeding back to the diesel:

Table 3.9.3

	DC-FT	kW	*Feed back*: (*Peak*) (kW)
a.	Lowering – decelerating	600 · 1,6 = 960	960 · 0,812 = 779
b.	Trolley travelling – deceleration	200 · 2 = 400	400 · 0,767 = 307
			1086

(The mechanical efficiency of the mechanism is reversed)

This results in feedback of 1086 kW.

The diesel can dissipate 10 percent of 1450 kW = 145 kW

To be dissipated by the flywheel or resistances = 941 kW

The energy, to be dissipated by the flywheel or resistances, will also apply if the crane is being driven by the maximum windforce, including the auxiliaries. This figure should always be checked.

The number of kiloWatts required of the diesel could be restricted somewhat by using a mechanism with a ‘constant power’ characteristic. In this case the factor f_a could be reduced, giving a somewhat smaller diesel output reduction. However in that case, the average acceleration and deceleration of the movement is also reduced, which influences the acceleration and deceleration time of the mechanism. Through this the throughput of the crane becomes less. Thus, if throughput is an important feature, no reduction should be made in the average acceleration and deceleration, otherwise the cycle time will lengthen, and throughput will be reduced.

Moreover, care must be taken as the PLC used in the crane can be influenced by quite a small dip in the diesel engine power output and even stop the crane.

(Use an UPS – uninterruptible power supply to prevent a dip of the PLC)

By fine tuning the diesel generator set, the time necessary for building up the diesel power (power response) to meet sudden demand, can be reduced somewhat. The main point is that the diesel generator set must be able to follow the fast and sudden changing load demands of the crane.

(b) The Alternating Current Frequency Control system

With the most modern alternating-current frequency control system (AC-Fr.C) the normal torque–speed diagram for the electric motor can be made as in Fig. 3.9.3. The current that the generator behind the

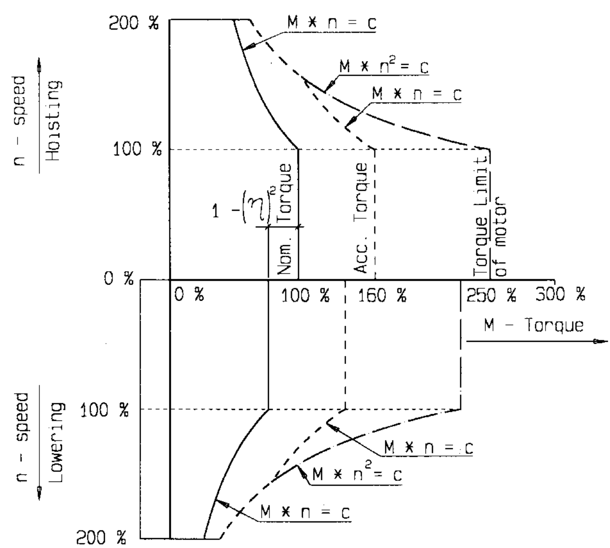


Fig. 3.9.3 AC frequency control: torque–speed diagram for hoisting/lowering

diesel has to deliver is proportional to the power (in kW) that the electric motor of the mechanism has to give.

- The cos-phi is approximately 0,95.
- AC-Fr.C asks approximately 0 percent current when $n = 0$ and $M = 160$ percent.
- The peak torque of the motor follows the curve $M \cdot n^2 = C$.
- The necessary power P in kW is the product of $n \cdot M$; just as it is in the Ward–Leonard–Krämer system.

Take:

$N_a = 1{,}25 \cdot$ power of the hoisting mechanism.

(The factor of 1,25 is used to produce some reserve power for acceleration, etc. The acceleration time then will be somewhat larger than with the diesel driven FT systems of Table 3.9.2.)

Total efficiency:

$$\eta_1 = 0{,}9 \cdot 0{,}95 \cdot 0{,}95 \cdot 1 = 0{,}812$$

$$\eta_2 = 0{,}9 \cdot 1 \cdot 0{,}95 \cdot 1 = 0{,}855$$

The possibilities of simultaneously working mechanisms are shown in Table 3.9.5.

Table 3.9.4

AC-Fr.C	*A*	*B*	*C*	*D*	*E*
	Hoisting mechanism	*Trolley travelling mechanism*	*Boom hoist mechanism*	*Crane travel mechanism*	*Auxiliaries (lighting, heating, air conditioning, maintenance crane, pumps)*
Number of kilowatts	600 kW $f_a = 1{,}25$	200 kW $f_a = 2$	250 kW $f_a = 1{,}25$	400 kW $f_a = 2$	80 kW
Efficiency of the motor(s) etc.	0,9	0,9	0,9	0,9	0,9
Efficiency between motor(s) and generator	0,95	0,95	0,95	0,95	1
Efficiency of the generator	0,95	0,95	0,95	0,95	0,95
Efficiency between generator and diesel	1 (or 0,9 if there is a gearbox between generator and diesel)				
	η_1	η_1	η_1	η_1	η_2

Table 3.9.5

		AC-Fr.C		kW	*Necessary diesel output* (kW)	
I	A.	Hoisting/accelerating		600	(600 · 1,25) : 0,812 =	923,6
	B	Auxiliaries		80	80 : 0,855 =	93,5
			Total	680	Diesel output =	1017
II	A.	Hoisting		600	(600 · 1) : 0,812 =	739
	B.	Trolley travelling/accelerating		200	(200 · 2) : 0,812 =	492,5
	E.	Auxiliaries		80	80 : 0,855 =	93,5
			Total	880	Diesel output =	1325
III	B.	Trolley travelling		200	(200 · 1) : 0,812 =	246
	D.	Crane travelling/accelerating		400	(400 · 2) : 0,812 =	985
	E.	Auxiliaries		80	80 : 0,855 =	93,5
			Total	680	Diesel output =	1325
IV	D.	Crane travelling (against storm)		400	(400 · 2) : 0,812 =	985,5
	E.	Auxiliaries		80	80 : 0,855 =	93,5
			Total	480	Diesel output =	1079

A diesel with an output of approximately 1325 kW Prime Power Rating can cope with this work. However, the acceleration time of the hoisting mechanism is somewhat increased. Care must be taken over the extra power losses of the diesel in high temperatures, the ability to follow the fast and sudden changing load demands of the crane and the feed back problems.

The crane builder should send the diesel generator manufacturer:

- a good calculation of the necessary Prime Power Rating;
- a cycle diagram of the crane, as well as the allowed voltage dip and frequency dip.

3.10 Calculating the power needed for the slewing motors of level luffing cranes

Factors to be considered are:

1. The resistance due to nominal slewing.
2. The resistance due to the influence of the wind.
3. The resistance due to the acceleration of the linear moving masses.
4. The resistance due to the acceleration of the rotating masses.

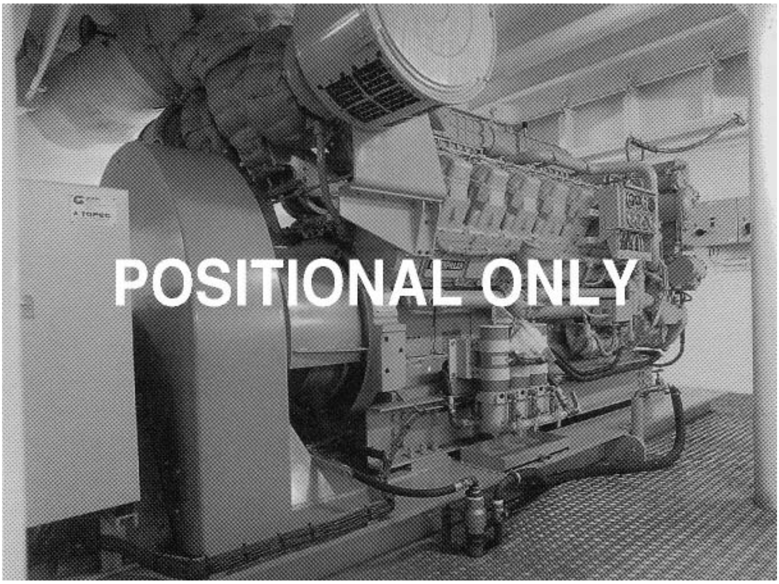


Fig. 3.9.4 Caterpillar diesel generator set with large flywheel

Main Characteristics

Slewing mechanism.

Maximum radius	m	$r_1 = 40$ m
Weight of the load	tons	$W_1 = 50$ t
Weight of the slewing part of the crane	tons	$W_t = 400$ t
Rev/min of the slewing motor	rev/min	$n = 1500$ rev/min
Speed of the crane	rev/min	$n = 1{,}2$ rev/min
Total efficiency of the slewing mechanism	η	$\eta = 0{,}9$
Type of crane		Double boom crane

Scheme of the slewing part:

(For simplification, the points where the centre of gravity of the weights and the points where the resultants of the wind load catch the crane, have been taken on the same radius.)

$W_1, W_2, \ldots$ = Weight of the different parts in tons.

$r_1, r_2, \ldots$ = Distance of the centre of gravity of the crane parts and the centre of the wind loads in m.

$F_1, F_2, \ldots$ = $c \cdot q \cdot F_1$; $c \cdot q \cdot \eta \cdot F_2$; $c \cdot q \cdot \eta \cdot F_3$; is the wind load on the different parts in kg; kN or t.

(See Section 1.5 for the calculation of c, q, η and F.)

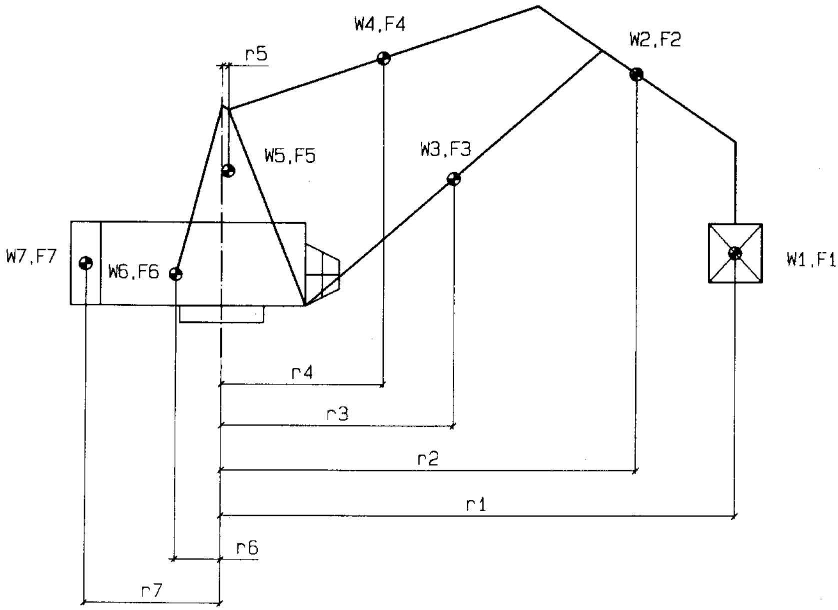


Fig. 3.10.1 Slewing luffing crane

1. The resistance due to nominal slewing

$$M_1 = (\Sigma W \cdot R) \cdot \mu \cdot 10 \qquad \text{kN m}$$

where

M_1 = resistance in kN m
ΣW = total weight in tons
R^1 = the resulting distance of the centre of gravity to the slewing axle in m
μ = the resistance of the slew bearing = 0,006

$W_1 = 50$ t	$r_1 = 40$ m	$W_1 \cdot r_1 = 2000$ tm
$W_2 = 10$ t	$r_2 = 30$ m	$W_2 \cdot r_2 = 300$ tm
$W_3 = 40$ t	$r_3 = 20$ m	$W_3 \cdot r_3 = 800$ tm
$W_4 = 16$ t	$r_4 = 16$ m	$W_4 \cdot r_4 = 256$ tm
$W_5 = 60$ t	$r_5 = 0$ m	$W_5 \cdot r_5 = 0$
$W_6 = 204$ t	$r_6 = -2$ m	$W_6 \cdot r_6 = -408$ tm
$W_7 = \underline{70\ \text{t}}$	$r_7 = -6$ m	$W_7 \cdot r_7 = \underline{-420\ \text{tm}}$

$\Sigma W = 450$ t $\qquad\qquad \Sigma W \cdot r = +2528$ tm

$$R^1 = \frac{\Sigma W \cdot r}{\Sigma W} \quad \text{m} \qquad\qquad R^1 = \frac{2528}{450}$$

= Centre of gravity of the whole upper crane $\qquad$ $= 5{,}61$ m

$$M_1 = 2528 \cdot 0{,}006 \cdot 10$$
$$= 151{,}6 \text{ kN m}$$

$$M_{1\,\text{motor}} = \frac{M_1}{i \cdot \eta}$$

$i = n_{\text{motor}}/n_{\text{crane}}$

$$i = \frac{1500}{1{,}2} = 1250$$

$$M_{1\,\text{motor}} = \frac{151{,}6 \cdot 10^3}{1250 \cdot 0{,}9} = 134{,}7 \text{ Nm} \qquad N_1 = \frac{M_1 \cdot n}{9550} = \frac{134{,}7 \cdot 1500}{9550} = 21{,}15 \text{ kW}$$

2. The resistance due to the influence of the wind

$M_2 = (\Sigma F \cdot r) \cdot 10$ $\qquad$ kN m

W (t)	r (m)	$F = c \cdot q \cdot \eta \cdot F$ (kg) (here: $q = 15$ kg/m^2)	$F \cdot r$ (tm)
$W_1 = 50$ t	$r_1 = 40$ m	$F_1 = c \cdot q \cdot \eta \cdot F_1 = 250$	$F_1 \cdot r_1 = 10$
$W_2 = 10$ t	$r_2 = 30$ m	$F_2 = c \cdot q \cdot \eta \cdot F_2 = 533$	$F_2 \cdot r_2 = 16$
$W_3 = 40$ t	$r_3 = 20$ m	$F_3 = c \cdot q \cdot \eta \cdot F_3 = 1500$	$F_3 \cdot r_3 = 30$
$W_4 = 16$ t	$r_4 = 16$ m	$F_4 = c \cdot q \cdot \eta \cdot F_4 = 700$	$F_4 \cdot r_4 = 11{,}2$
$W_5 = 60$ t	$r_5 = 0$ m	$F_5 = c \cdot q \cdot \eta \cdot F_5 = 3000$	$F_5 \cdot r_5 = 0$
$W_6 = 204$ t	$r_6 = -2$ m	$F_6 = c \cdot q \cdot \eta \cdot F_6 = 1600$	$F_6 \cdot r_6 = -3{,}2$
$W_7 = 70$ t	$r_7 = -6$ m	$F_7 = c \cdot q \cdot \eta \cdot F_7 = \underline{150}$	$F_7 \cdot r_7 = \underline{-0{,}9}$
		$\Sigma F = 7800$ kg	$\Sigma F \cdot r = 63{,}1$ tm

$$M_{2\,\text{motor}} = \frac{M_2 \cdot 10^3}{i \cdot \eta}\ \text{Nm} \qquad M_2 = 63{,}1 \cdot 10 = 631\ \text{kN m}$$

$$M_{2\,\text{motor}} = \frac{631 \cdot 10^3}{1250 \cdot 0{,}9} = 561\ \text{Nm}$$

$$N_2 = \frac{M_{2\,\text{motor}} \cdot n}{9550}\ \text{kW} \qquad N_2 = \frac{561 \cdot 1500}{9550} = 88\ \text{kW}$$

3. The resistance due to the acceleration of the linear moving masses.

$M_3 = \theta \Sigma T \cdot 10$ kN m

T = the inertia movement of a certain part, regarded to be the centre of rotation of the upper crane.

Simplified:
'Point' – (load) $T = m_1 \cdot r_1^2$
'Line' – (r from centre of gravity) $T = \frac{1}{12} \cdot m_2 \cdot L_2^2 + m_2 r_2^2$

T in ton metre/sec^2

θ = acceleration
(For heavy duty cranes
t_a = 6–8 *sec*) *in radii/sec*^{2}

See Fig. 3.10.2

W(t)	$m = (\frac{1}{10} W)$	r(m)	L(m)	T = (ton metre/sec^2)
$W_1 = 50$ t	$m_1 = 5$	$r_1 = 40$ m	–	$T_1 = m \cdot r^2 = 8000$
$W_2 = 10$ t	$m_2 = 1$	$r_2 = 30$ m	14	$T_2 = \frac{1}{12} \cdot m \cdot L^2 + m \cdot r^2 = 916$
$W_3 = 40$ t	$m_3 = 4$	$r_3 = 20$ m	35	$T_3 = \frac{1}{12} \cdot m \cdot L^2 + m \cdot r^2 = 2008$
$W_4 = 16$ t	$m_4 = 1{,}6$	$r_4 = 16$ m	29	$T_4 = \frac{1}{12} \cdot m \cdot L^2 + m \cdot r^2 = 522$
$W_5 = 60$ t	$m_5 = 6$	$r_5 = 0$ m	17	$T_5 = \frac{1}{12} \cdot m \cdot L^2 + m \cdot r^2 = 144$
$W_6 = 204$ t	$m_6 = 20{,}4$	$r_6 = -2$ m	17	$T_6 = \frac{1}{12} \cdot m \cdot L^2 + m \cdot r^2 = 572$
$W_7 = 70$ t	$m_7 = 7$	$r_7 = -6$ m	2	$T_7 = \frac{1}{12} \cdot m \cdot L^2 + m \cdot r^2 = 252$
				$\Sigma T = 12\,414$

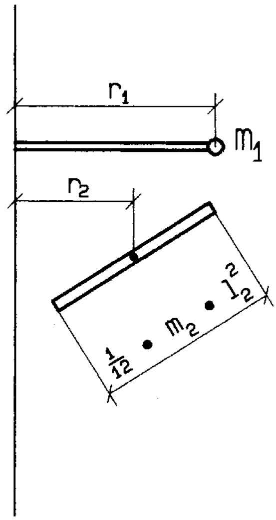


Fig. 3.10.2 Inertia scheme

Take $t_a = 8$ sec

$\theta = \dfrac{2 \cdot \pi \cdot n}{60 \cdot t_a}$ rad/sec^2 $\theta = \dfrac{2 \cdot \pi \cdot 1{,}2}{60 \cdot 8}$

$= 0{,}0157$

$M_3 = \theta \Sigma T \cdot 10$ kN m $M_3 = 0{,}0157$

$\cdot\, 12\,414 \cdot 10$

$= 1949$ kN m

$$M_{3\,\text{motor}} = \frac{M_3 \cdot 10^3}{i \cdot \eta}$$

$$= \frac{1949 \cdot 10^3}{1250 \cdot 0{,}9}$$

$= 1732$ Nm

$$N_3 = \frac{M_{3\,\text{motor}} \cdot n}{9550}\ \text{kW}$$

$$= \frac{1732 \cdot 1500}{9550}$$

$= 272$ kW

4. The resistance due to the acceleration of the rotating masses:

$J = \Sigma$ mom. of inertia of the rotating parts	kg m^2	$J = 6$ kg m^2
n_m = rev. of min of the motor	rev/min	
$\omega = \dfrac{n_m \cdot 2 \cdot \pi}{60}$	rad/sec	$\omega = \dfrac{1500 \cdot 2 \cdot \pi}{60} = 157$ Nm
$M_a = \dfrac{J \cdot \omega}{t_a}$	Nm	$M_4 = \dfrac{6 \cdot 157}{8} = 118$ Nm
$N_4 = \dfrac{M_4 \cdot n_m}{9550}$	kW	$N_4 = \dfrac{118 \cdot 1500}{9550} = 18{,}5$ kW

Addition:

1.	Nominal slewing	$N_1 = 21{,}15$ kW
2.	Wind, $q = 150$ N/m^2	$N_2 = 88$ kW
	Total for normal slewing	$\Sigma N = 109{,}15$ kW

During acceleration:

1.	Nominal slewing	$N_1 = 21{,}15$ kW
2.	Wind, $q = 150$ N/m^2	$N_2 = 88$ kW
3.	Acceleration of the linear moving masses, $t_a = 8$ sec	$N_3 = 272$ kW
4.	Acceleration of the rotating masses	$N_4 = 18{,}5$ kW
	Total during acceleration	$\Sigma N = 399{,}65$ kW

The motor power needed must now be greater than $\Sigma N = 109{,}15\,\text{kW}$ and $\Sigma N = 399{,}65 : f_a\,\text{kW}$

Take $f_a = 2$ ($M_{max} = 200$ percent of M_{nom}).

So, ΣN must be greater than $\Sigma N = 110\,\text{kW}$ and ΣN must be greater than $\Sigma N = 399{,}65/2 = 199{,}8\,\text{kW}$.

Take 4 slewing mechanisms for this crane, each 50 kW; $n = 1500$; $f_a = 2$. $S_3 - 60$ percent rating.

(If a slewing crane is positioned on a pontoon, the influence of the slewing against the slope of the pontoon also has to be calculated.)

3.11 Calculating the power needed for the luffing motor of level luffing cranes

Although there are a number of level luffing systems, e.g. hydraulic cylinders; pin-and-rack; tooth segments; etc., which can be used as luffing mechanisms, only the system which uses a tackle is considered here. All other systems can be easily derived from this tackle-system.

Factors to be considered are:

1. The resistance due to nominal luffing.
2. The resistance due to the influence of the wind.
3. The resistance due to the acceleration of the linear moving masses.
4. The resistance due to the acceleration of the rotating masses.

Main characteristics: level luffing mechanism

Maximum outreach from centre of jib hinge point	m	$r_1 = 36\,\text{m}$
Minimum outreach from centre of jib hinge point	m	$r_2 = 12\,\text{m}$
Weight of the load	tons	$W_1 = 40\,\text{t}$
Weight of the jib	tons	$W_2 = 25\,\text{t}$
Horizontal level luffing speed	m/min	$v = 60\,\text{m/min}$
n_{motor}	rev/min	$n = 1500\,\text{rev/min}$
Total efficiency (of tackle and gearings)	η	$\eta = 0{,}9$ single jib crane; heavy duty

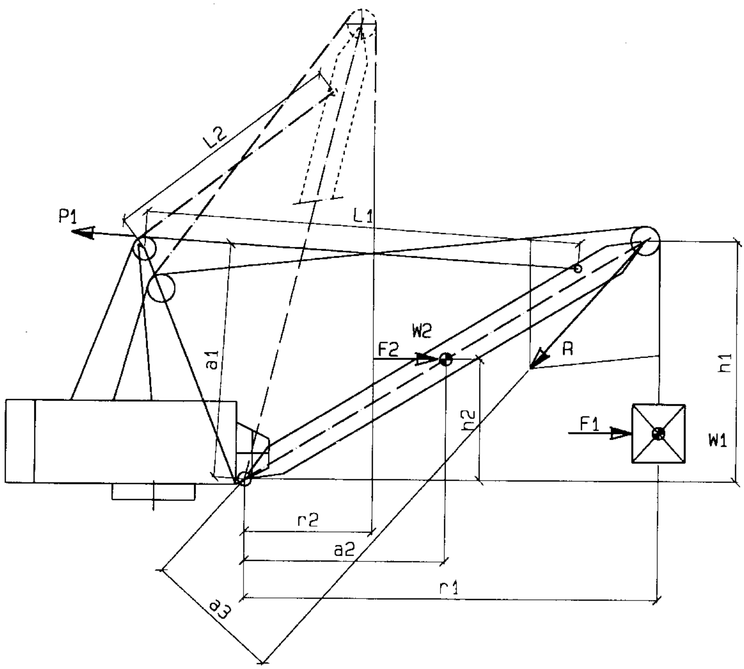


Fig. 3.11.1 Level luffing system

(For simplification the points where the centre of gravity of the weights, and the points where the resultant of the windloads catch the crane, have been taken as the same points. Horizontal forces on the load (wind load; centrifugal forces out of slewing) attack the cranejib in the centre of the ropesheave(s) in the top of the jib.

$W_1, W_2, \ldots$	= Weight of the different parts in tons
R	= The resultant of the wirerope forces out of the load W_1; acting on the jib, in tons
$r_1, r_2; a_1, a_2;$ $L_1, L_2; h_1, h_2$	= Distances in m
P_1	= Ropeforce in tons
F_1, F_2	$= c \cdot q \cdot F_a$; $c \cdot q \cdot F_b$; is the windload on the different parts in kg; kN or t. (See for example Section 1.5 for the calculation of c, q, and F.)

Example

1. The resistance due to nominal luffing:

$M_1 = (R \cdot a_3 + W_2 \cdot a_2) \cdot 10$	kN m	$M_1 = (47 \cdot 1 + 25 \cdot 16) \cdot 10$ $= 4470$ kN m
M_1 = resistance in the centre of rotation of the jib; in kN m		

2. The resistance due to the influence of the wind:

$M_2 = (F_1 h_1 + F_2 \cdot h_2) \cdot 10$	kN m	
$F_1 = c \cdot q \cdot F_a = 1$ t	$h_1 = 24$ m	$F_1 \cdot h_1 = 24$ tm
$F_2 = c \cdot q \cdot \eta \cdot F_b = 1{,}5$ t	$h_2 = 11$ m	$F_2 \cdot h_2 \cdot 16{,}5$ tm $M_2 = (24 + 16{,}5) \cdot 10$ $= 405$ kN m
The force in the tackle due to M_1 and M_2 is:		
$P_1 = \dfrac{M_1 + M_2}{a_1}$	kN	$P_1 = \dfrac{4470 + 405}{26{,}25} = 186$ kN
The speed in the tackle can be derived as follows:		
Horizontal level luffing speed		$v = 60$ m/min $v_1 = 1$ m/sec
Distance to be covered $= r_1 - r_2$	m	$r_1 - r_2 = 36 - 12 = 24$ m
Nos. of seconds for luffing:	sec	$t = \dfrac{r_1 - r_2}{v_1} = \dfrac{36 - 12}{1}$ $= 24$ sec
Hauling speed of the tackle:		
$v_t = \dfrac{l_1 - l_2}{t}$	m/sec	$v_t = \dfrac{36 - 15{,}75}{24}$ $= 0{,}84$ m/sec

Power needed: $N = \dfrac{P_1 \cdot v_t}{\eta}$ kW $\quad N = \dfrac{186 \cdot 0{,}84}{0{,}9}$

$= 174\,\text{kW}$

3. The resistance due to the acceleration of the linear moving masses:

Take $t_a = 4$ sec

$F = m \cdot a$

$F = \dfrac{P_1}{9{,}81} \cdot \dfrac{v_t}{t_a}$ kN $\quad F = \dfrac{186}{9{,}81} \cdot \dfrac{0{,}84}{4}$

$= 4\,\text{kN}$

$N_3 = \dfrac{F \cdot v}{\eta}$ kW $\quad N_3 = \dfrac{F \cdot v}{\eta}$

$= \dfrac{4 \cdot 0{,}84}{0{,}9}$

$= 3{,}73\,\text{kW}$

4. The resistance due to the acceleration of the rotating masses:

$J = \Sigma_{\text{mom}}$ of inertia of the rotating parts $\quad$ kgm^2

n_m = min rev. of the motor $\quad$ rev/min

$\omega = \dfrac{n_m \cdot 2 \cdot \pi}{60}$ rad/sec $\quad \omega = \dfrac{1500 \cdot 2 \cdot \pi}{60}$

$= 157$ rad/sec.

$J = 9\,\text{kgm}^2$

$M_4 = \dfrac{J \cdot \omega}{t_a}$ Nm $\quad M_4 = \dfrac{9 \cdot 157}{4}$

$= 353\,\text{Nm}$

$N_4 = \dfrac{M_4 \cdot n_m}{9550} \quad N_4 = \dfrac{353 \cdot 1500}{9550}$

$= 55{,}4\,\text{kW}$

Conclusion

As can be seen from 1. and 2., the needed motor power out of 1. and 2. is $N = 174\,\mathrm{kW}$.

The influence of acceleration 3. and 4. is only $N = 3{,}73 + 55{,}4 = 59{,}13\,\mathrm{kW}$.

Luffing-in is critical; luffing-out asks for less power ($\times\,\eta^2$) but asks for power, full electric braking and powerful mechanical brakes.

The force P_1 in the tackle diminishes rapidly when luffing-in and increases rapidly when luffing-out.

Take motor:

$N = 170\,\mathrm{kW}$

$n = 1500\,\mathrm{rev/min}$

$S_3 - 60$ percent rating

$f_a = 160$ percent.

If a level luffing crane is positioned on a pontoon, the influence of the luffing against the slope of the pontoon also has to be calculated.

In the calculation the swinging of the load is neglected.

For fast working grabbing cranes especially, the influence of the swinging of the load on the luffing mechanism and the luffing motor should be calculated.

Chapter 4

Brakes

4.1 Modern brakes

Sections 4.2 to 4.5 give the calculations of the hoisting brakes. Section 4.7 shows how the braking distance of a crane can be calculated.

Modern brake types have a brake torque which is normally calculated with a friction coefficient of $\mu = 0{,}4$ and they must give a brake torque of $1{,}6 \times$ the nominal motor torque up to $2{,}2 \times$ the nominal motor torque. For the braking of 'vertical movements', e.g. hoisting/lowering or lowering/hoisting the boom normally disc brakes are used.

These disc brakes have the advantage that the inertia movement of the brake disc is low, compared to that of the brake drum of a drum brake. Besides:

- One brake size may be used for different disc diameters, while a drum brake requires a different brake size for every different drum diameter.
- The friction material available for modern disc brakes allows for far higher operational speeds and temperatures of the brake lining, thus offering a high level integrity in case of emergency.
- The friction properties of non-organic disc brake friction materials are much less influenced by corrosion, pollution, and humidity.
- The life cycle of disc braking linings compared to drum brake linings – providing they are used for the same application – is 50 to 100 percent longer.

A number of specialized manufacturers sell excellent brakes. The examples shown in Tables 4.1.1 to 4.1.3 show figures from the well-known manufacturers of brakes, Bubenzer of Kirchen in Germany. Bubenzer provides the following figures for their disc brakes.

Table 4.1.1

Brake type SB 14.11							*J brake disc+ coupling (kgm^2)*
Thrustor type					Ed 23/5	Ed 30/5	
Contact load in N					2500	3400	
Brake disc	Brake torque $M_{Br.Max}$ in Nm at an average friction of $\mu = 0{,}4$						
250					200	270	
280					230	310	
315					260	355	
355					300	410	0,4
400					345	470	0,6
450					395	540	1,0
500					445	610	1,5

The drum brakes are sometimes somewhat cheaper than the disc brakes; however the brake torque can be less than that of the disc brake of the same diameter.

Table 4.1.2

Brake type SB 23							*J brake disc+ coupling (kgm^2)*
Thrustor type			Ed 50/6	Ed 80/6			
Contact load in N			8500	12 500			
Brake disc	Brake torque $M_{Br.Max}$ in Nm at an average friction of $\mu = 0{,}4$						
355			935				
400			1085	1600			
450			1255	1850			
500			1425	2100			2
560			1630	2400			3
630			1870	2750			4,5

Table 4.1.3

Brake type SB 28 S						*J brake disc+ coupling (kgm²)*
Thrustor type		Ed 121/6	Ed 121/6	Ed 301/6	Ed 301/8	
Brake disc	Brake torque $M_{Br.Max}$ in Nm at an average friction of $\mu = 0{,}4$					
500		3100	5040			2
560		3550	5750			3
630		4100	6600	9700	12 500	4,5
710		4700	7600	11 200	14 400	7,5
800		5400	8800	12 800	16 500	11
900				14 700	18 900	15
1000				16 500	21 200	22

Trolley-travelling and crane-travelling mechanisms often use a plate brake. This is in fact a disc brake. These brakes are built-in in the motors on the second motorshaft. These brakes are not so easy to control and adjust. The brake torque can show a tolerance of plus or minus 15 percent which makes it difficult to predict the exact brake torque which the plate brake will give.

With a crane or trolley running at a high speed, it is wise to calculate the brake distance of the crane or trolley in accordance with Section 4.7. With trolley and crane travel brakes, which can easily be adjusted, a disc brake or a drum brake can be used, released by a thrustor or a DC solenoid. The brake torque of such brakes should be taken as approximately 1,5 to 1,8 times the nominal motor torque.

With boom-hoist mechanisms it is usual to build in a disc brake or drum brake between the motor and the in-going shaft of the gearbox. Additionally, a second, so-called emergency brake is installed on the boom-hoist drum. Should something happen during lowering the boom and if the motor speed reaches 110 percent of the nominal motor speed, a centrifugal switch activates this emergency brake.

The emergency brake used to be constructed as a bandbrake, now, caliper disc brakes are normally used. However, some crane users still prefer a bandbrake, because the brake rim under a bandbrake becomes less corroded than the brake sheave of a disc brake. To avoid rust and corrosion on the disc surface it is possible to provide the caliper disc

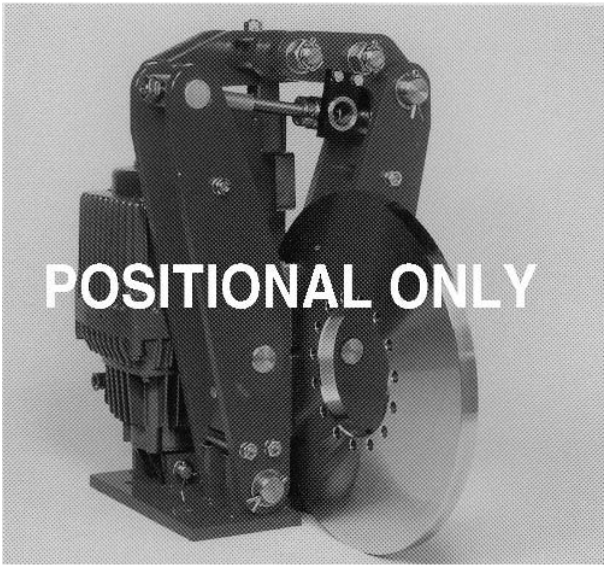


Fig. 4.1.1 Disc brake

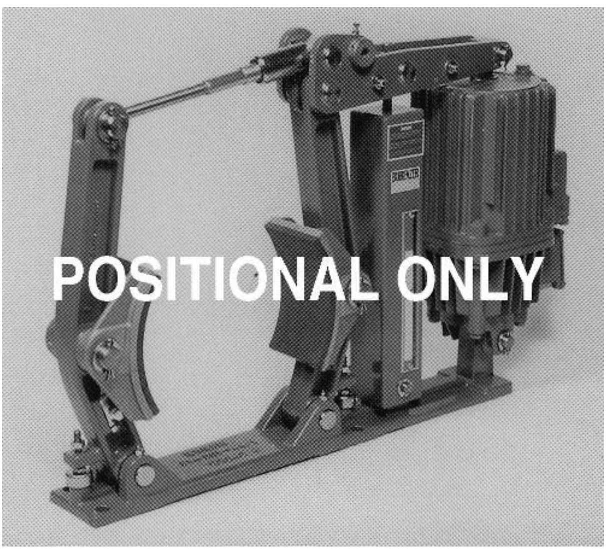


Fig. 4.1.2 Drum brake

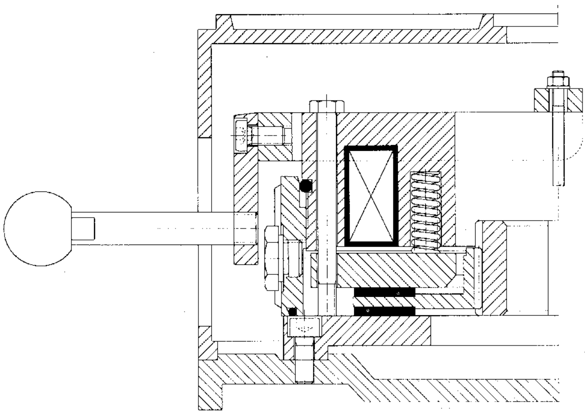


Fig. 4.1.3 Plate brake

brake with cleaning pads. Bubenzer mention 4 caliper disc brake types – see Table 4.1.4. To give a hoisting winch extra safety, in the case of a severe breakdown in the gearbox, some crane users demand an extra disc- or bandbrake on the hoist drum itself. In case of such a calamity in the gear box, these brakes must then catch the load.

Maintenance

The maintenance period of the brakes are described in Chapter 10 Maintenance. It is clear that the brakes have to be inspected, controlled, and maintained very regularly, because of the enormous problems caused if a brake should fail.

The monitoring of disc brakes

The preventative maintenance of disc brakes can be simplified by installing a monitoring system. When a number of sensors are installed on

Table 4.1.4 Caliper disc brakes

Type	*SF10*	*SF15*	*SF24*	*SF40*
Contact load	100 KN	150 KN	240 KN	400 KN
Friction coefficient	$\mu = 0{,}4$	$\mu = 0{,}4$	$\mu = 0{,}4$	$\mu = 0{,}4$
Operating factor	$f = 40$	$f = 60$	$f = 96$	$f = 160$
Operating diameter	d (mm)	d (mm)	d (mm)	d (mm)
Brake torque in Nm		$M_{br} = f \cdot d$ (Nm)		

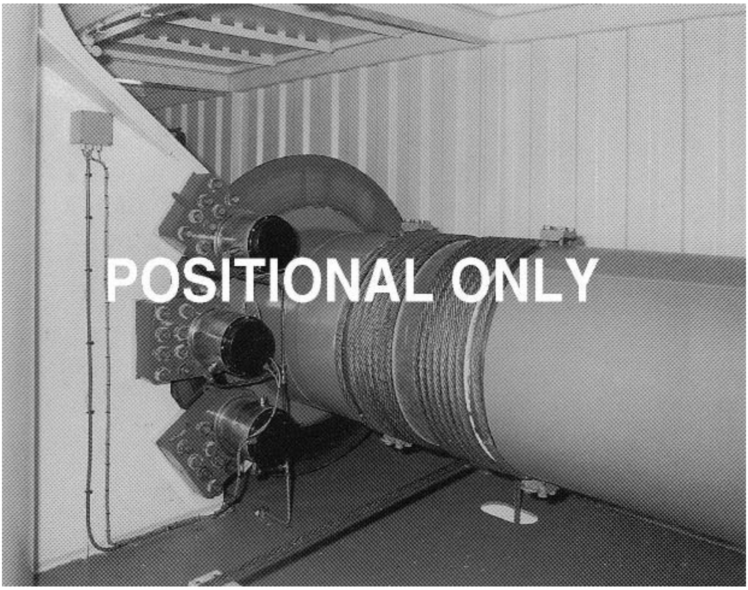


Fig. 4.1.4 Calliper disc brake

each brake the following items can be monitored:

- the contact pressure of each brake pad, giving an indication of the brake torque;
- the brake pad temperature;
- the brake pad wear;
- the brake disc speed;
- the position of the thrustor.

Data from these sensors are submitted and visualized via a field bus system to the main control system or via radio signals to a separate computer or modem. Bubenzer can install this CCB (Computer Controlled Brake) system on a series of disc brakes in one or more cranes. Permanent monitoring of brake systems and print outs of brake reports can even be produced from the central maintenance room where further maintenance then can be organized, etc.

4.2 Hoisting brakes

Lowering the load; emergency stop

This can be a dangerous situation. A crane is lowering a load; and an emergency stop is required – the crane driver then pushes the emergency button. The hoisting mechanism does not brake electrically: it is an emergency stop. The load is now pulled down by gravity and accelerated sharply during the short time that it is needed to activate the brake. The activated brake starts braking the load, but is starting from a higher speed than the normal lowering speed. This means that the necessary time for braking towards zero-speed becomes longer than normal. Assuming that it is a container crane with the reeving shown in Fig. 4.2.1 (schematic).

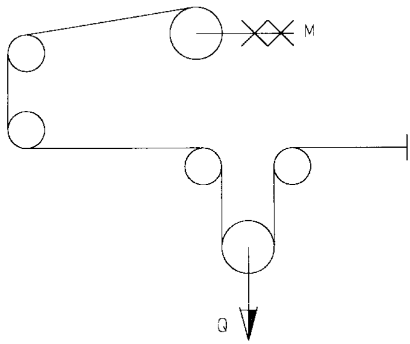


Fig. 4.2.1 Wire rope scheme

Example

1. Weight of spreader plus load: Q (kg) — $Q = 66\,000$ kg
2. Force on the wire ropes on the drum: L (kg) (see Fig. 4.2.1)
   $L = (Q:2) \cdot \eta_s$ (kg) — $L = (66\,000:2) \cdot 0{,}95 = 31\,350$ kg
   $\eta_s = \eta_{\text{sheaves + drum}} = n \cdot 0{,}99$ — $(n = 5)$
3. Torque on motorshaft:
   $$M_1 = \left(L \cdot \frac{D_{\text{drum}}}{2} : i_{\text{gb}}\right) \cdot \eta_{\text{gb}} \cdot 9{,}81 \text{(Nm)}$$
   $$M_1 = \left(31\,350 \cdot \frac{1{,}2}{2} : 24{,}6\right) \cdot 0{,}94 \cdot 9{,}81 = 7051 \text{ Nm}$$
   Drum diam. $D_{\text{drum}} = 1{,}2$ m
   Gearbox reduction $i_{\text{gb}} = 24{,}6$
   Efficiency gearbox $\eta_{\text{gb}} = 0{,}98^3$

4. Lowering speed of the load: v (m/min) — $v = 60$ m/min

5. Wire rope speed on the drum:

$v_d = 2 \cdot v$ (m/min) — $v_d = 2 \cdot 60 = 120$ m/min

$= (2 \cdot v) : 60$ (m/sec) — $= 120 : 60 = 2$ m/sec

6. Nos. of rev/min of the drum:

$n_d = v_d : (\pi \cdot D_d)$ (rev/min) — $n_d = 120 : (\pi \cdot 1{,}2)$

$= 31{,}84$ rev/min

7. Nos. of rev/min of the motor:

$n_{motor} = n_d \cdot i_{gb}$ (rev/min) — $n_m = 31{,}84 \cdot 24{,}6$

$= 783$ rev/min

8. Inertia moment on the motorshaft from: motor(s); brakesheave(s) and gearbox:

$J_{rot} = J_m + J_b + J_{gb}$ (kg m^2) — $J_{rot} = 24 + 16 + 6$

$= 46$ kg m^2

9. Reduced inertia moment on the motorshaft from the weight of the spreader plus load:

$J_L = (L \cdot v_d^2 \cdot \eta_{gb}) : \omega_1^2$ (kg m^2) — $J_L = (31\,350 \cdot 2^2 \cdot 0{,}94) : \left(\dfrac{783 \cdot 2 \cdot \pi}{60}\right)^2$

$J_L = 117\,876 : 81{,}95^2$

$= 17{,}55$ kg m^2

10. $J_{total} = J_{rot} + J_L$ (kg m^2) — $J_{tot} = 46 + 17{,}55$

$= 63{,}55$ kg m^2

11. After pushing the emergency button, the load is accelerated by M_1 during Δt sec (activating time for the brake) with $\Delta\omega_2$ (rad/sec):

$$\Delta\omega_2 = \frac{\Delta t \cdot M_2}{J_{\text{rot}}} \text{(rad/sec)} \qquad \Delta\omega_2 = \frac{(0{,}3 \cdot 7051)}{46} = 46 \text{ rad/sec}$$

12. The activated brake starts mechanical braking after Δt sec with a rotational speed on the motorshaft of:

$$\omega_3 = (\omega_1 + \omega_2)\text{(rad/sec)} \qquad \omega_3 = \left(\frac{783 \cdot 2\pi}{60}\right) + 46$$

$$\omega_1 = (n_{\text{m}} : 60) \cdot 2\pi \text{ (rad/sec)} \qquad = 81{,}95 + 46$$

$$= 127{,}95 \text{ rad/sec}$$

13. The numbers of rev/min of the motor- and brake-shaft is then:

$$n_2 = \omega_3 \cdot \frac{60}{2\pi} \text{(rev/min)} \qquad n_2 = 128 \cdot \frac{60}{2\pi} = 1223 \text{ rev/min}$$

14. The wire rope speed on the drum is then:

$$v_{\text{d2}} = \frac{n_2}{n_{\text{m}}} \cdot v_{\text{d}} \text{ (m/sec)} \qquad v_{\text{d2}} = \frac{1223}{783} \cdot 2 = 3{,}123 \text{ m/sec}$$

15. The effective brake moment is:

$$M_{\text{be}} = \eta_{\text{b}} \cdot M_{\text{b}} \text{ (Nm)} \qquad M_{\text{be}} = 0{,}95 \cdot 19\,000$$

$$= 18\,050 \text{ Nm}$$

16. The effective braking time is:

$$t_{\text{brake}} = \frac{\omega_3 \cdot J_{\text{total}}}{M_{\text{be}} - M_1} \text{(sec)} \qquad t_{\text{br}} = \frac{128 \cdot 63{,}55}{18\,050 - 7051} = 0{,}740 \text{ sec}$$

17. Total braking time:

$$t = (\Delta t + t_{\text{br}}) \text{ (sec)} \qquad t = 0{,}3 + 0{,}74 = 1{,}04 \text{ sec}$$

18. Wire rope displacement on the drum during braking:

$S_d = \Delta S_1 + \Delta S_2$ (m)

ΔS_1 = displacement on the drum during Δt (m)

ΔS_2 = displacement on the drum during deceleration whilst braking in t_{br} sec.

$$\Delta S_1 = \frac{v_d + v_{d2}}{2} \cdot \Delta t \text{ (m)} \qquad \Delta S_1 = \frac{2 + 3{,}123}{2} \cdot 0{,}3 = 0{,}768\text{ m}$$

$$\Delta S_2 = \frac{v_{d2}}{2} \cdot t_{br} \text{ (m)} \qquad \Delta S_2 = \frac{3{,}123}{2} \cdot 0{,}74 = 1{,}155\text{ m}$$

$$S_d = 0{,}768 + 1{,}155 = 1{,}923\text{ m}$$

19. Total displacement of spreader and load during emergency stop: in lowering direction (see Fig. 4.2.2)

$$S_{spr+L} = S_d : 2 \text{ (m)} \qquad S_{spr+L} = 1{,}923 : 2 = 0{,}96\text{ m}$$

Note:

- The winch has: 2 motors;
  2 brakes;
  1 gearbox;
  2 drums.
- The spreader and the load are suspended by eight ropefalls; 4 of which are fixed on the boom end; the other 4 falls run via wire rope sheaves to the 2 wire rope drums.
- Δt can be taken as $\Delta t = 0{,}3$ sec
  the worst case for $\Delta t = 0{,}5$ sec.
- The maximal peripheral speed of the brake disc must be controlled.

The allowed number of brake cycles in an emergency stop can be calculated as follows:

Dissipated energy per brake cycle: $$\text{WB} = M_{br} \cdot \frac{n_2}{9{,}55} \cdot \frac{t_{br}}{2000} \text{ (kJ)}$$

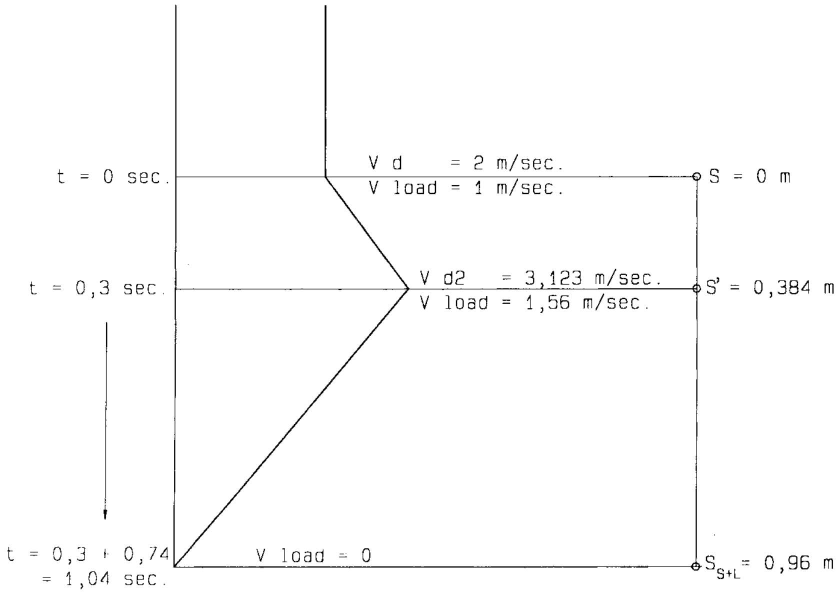


Fig. 4.2.2 Lowering: emergency stop

Nos. of kWh per brake cycle: $$\mathrm{kWh} = \frac{\mathrm{kJ}}{3593}\ (\mathrm{kWh})$$

Allowed numbers of emergency brake cycles; approximately: $$Z = \frac{v_B \cdot 2}{\mathrm{kWh} \cdot 30}\ (\mathrm{nos})$$

where

$v_B = 98\,100\ \mathrm{mm}^3$ for SB23 brakes (for a certain brakepad material)

$v_B = 244\,800\ \mathrm{mm}^3$ for SB28 brakes (for a certain brakepad material)

30 = brakepad wear per kWh.

The maximum circumference speed of the brake disc which is allowed is: $v = 85\ \mathrm{m/sec}$ for a brake disc of Fe52.3 (S 355 J2 G3).

4.3 Hoisting brakes

Lowering the load; braking by full motor torque

The crane driver is lowering the load and wants to halt the load by stopping the winch by ‘electric braking’. The full motor torque is to be

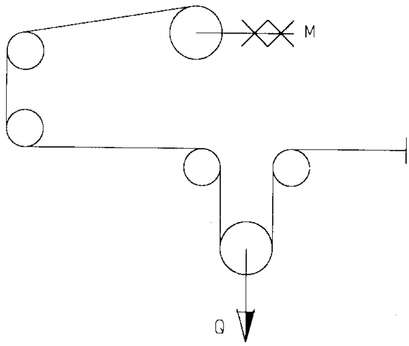


Fig. 4.3.1 Wire rope scheme

taken as the brake moment. The reeving scheme in Fig. 4.3.1 is assumed for a container crane.

Example

1. Weight of spreader plus Load: Q (kg) — $Q = 66\,000$ kg

2. Force on the wire ropes on the drum:

   L (kg) (see Fig. 4.3.1)

   $L = (Q:2) \cdot \eta_s$ (kg) — $L = (66\,000:2) \cdot 0{,}95 = 31\,350$ kg

   $\eta_s = \eta_{\text{sheaves + drum}} = n \cdot 0{,}99$ — $(n = 5)$

3. Torque on motorshaft:

   $$M_1 = \left(L \cdot \frac{D_{\text{drum}}}{2} : i_{gb}\right) \cdot \eta_{gb} \cdot 9{,}81 \text{ (Nm)}$$

   $$M_1 = \left(31\,350 \cdot \frac{1{,}2}{2} : 24{,}6\right) \cdot 0{,}94 \cdot 9{,}81 = 7051 \text{ Nm}$$

   Drum diam. $D_{\text{drum}} = 1{,}2$ m

   Gearbox reduction $i_{gb} = 24{,}6$

   Efficiency gearbox $\eta_{gb} = 0{,}98^3$

4. Lowering speed of the load:

v (m/min) $\quad$ $v = 60$ m/min

5. Wire rope speed on the drum:

$v_d = 2 \cdot v$ (m/min) $\quad$ $v_d = 2 \cdot 60 = 120$ m/min

$= (2 \cdot v) : 60$ (m/sec) $\quad$ $= 120 : 60 = 2$ m/sec

6. Nos. of rev/min of the drum:

$n_d = v_d : (\pi \cdot D_d)$ (rev/min) $\quad$ $n_d = 120 : (\pi \cdot 1{,}2)$

$= 31{,}84$ rev/min

7. Nos. of rev/min of the motor:

$n_{motor} = n_d \cdot i_{gb}$ (rev/min) $\quad$ $n_m = 31{,}84 \cdot 24{,}6$

$= 783$ rev/min

8. Inertia moment on the motorshaft from:

Motor(s); brake sheave(s) and gearbox:

$J_{rot} = J_m + J_b + J_{gb}$ (kg m^2) $\quad$ $J_{rot} = 24 + 16 + 6$

$= 46$ kg m^2

9. Reduced inertia moment on the motorshaft from the weight of the spreader plus load:

$J_L = (L \cdot v_d^2 \cdot \eta_{gb}) : \omega_1^2$ (kg m^2) $\quad$ $J_L = (31\,350 \cdot 2^2 \cdot 0{,}94) : \left(\frac{783{,}2\pi}{60}\right)^2$

$J_L = 117\,876 : 81{,}95^2$

$= 17{,}55$ kg m^2

10. $J_{total} = J_{rot} + J_L$ (kg m^2) $\quad$ $J_{tot} = 46 + 17{,}55$

$= 63{,}55$ kg m^2

11. Braking is immediately started with the electric current, delivering the nominal motor torque. (The 2 motors deliver in total

$N = 720\,\text{kW}$ at

$n = 783$ rev/min)

$$M_{\text{nom}} = \frac{N \cdot 9550}{n}\ (\text{Nm}) \qquad M_{\text{nom}} = \frac{720 \cdot 9550}{783}$$

$$M_{\text{nom}} = M_{\text{el.braking torque}} \qquad M_{\text{nom}} = M_{\text{eb}} = 8781\ \text{Nm}$$

12. $$\omega_{\text{mot}} = \frac{n}{60} \cdot 2\pi\ (\text{rad/sec}) \qquad \omega_{\text{mot}} = \frac{783}{60} \cdot 2\pi\ (\text{rad/sec})$$

$$= 81{,}95\ \text{rad/sec}$$

13. The effective braking time is:

$$t_{\text{brake}} = \frac{\omega_{\text{mot}} \cdot J_{\text{total}}}{M_{\text{eb}} - M_1}\ (\text{sec}) \qquad t_{\text{br}} = \frac{81{,}95 \cdot 63{,}55}{8781 - 7051}\ \text{sec} = 3\ \text{sec}$$

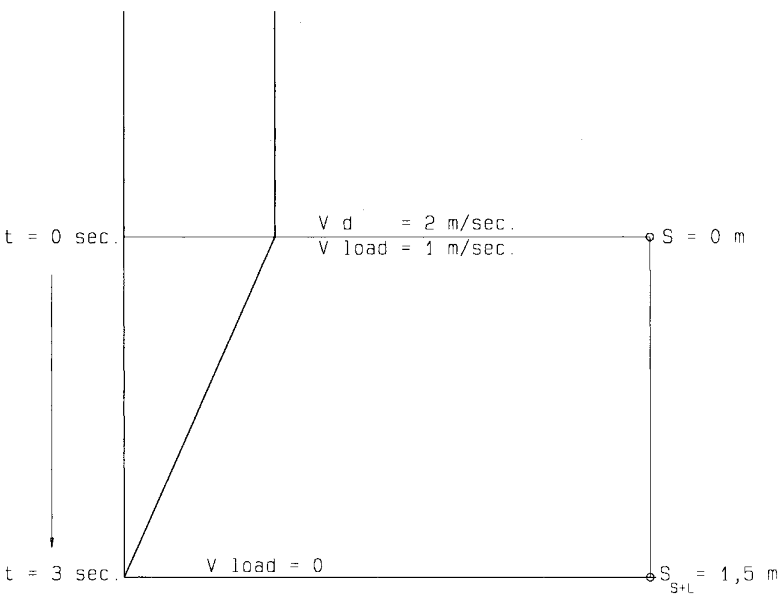


Fig. 4.3.2 Lowering: electrical braking by full motor torque

14. Wire rope displacement on the drum during braking:

$S_d = \frac{1}{2} \cdot v_d \cdot t_{br}$ (m) $\qquad S_d = \frac{1}{2} \cdot 2 \cdot 3 = 3$ m

15. Total displacement of spreader and load during electric braking in hoisting direction:

$S_{spr+L} = S_d : 2$ (m) $\qquad S_{spr+L} = 3 : 2 = 1{,}5$ m

4.4 Hoisting brakes

Hoisting the load; braking by full motor torque

The crane driver is hoisting the load and wants to stop the load by stopping the hoisting winch by 'electric braking'. We take now the full motor torque as brake moment. Consider a container crane with the reeving shown in Fig. 4.4.1 (schematic).

Example

1. Weight of spreader plus load: Q (kg) $\qquad Q = 66\,000$ kg

2. Force on the wire ropes on the drum:

L (kg) (see Fig. 4.4.1)

$L = (Q:2) \cdot \eta_s$ (kg) $\qquad L = (66\,000:2) \cdot 0{,}95$

$= 31\,350$ kg

$\eta_s = \eta_{sheaves+drum} = n \cdot 0{,}99 \qquad (n = 5)$

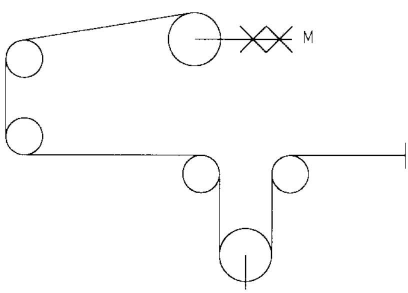


Fig. 4.4.1

3. Torque on motorshaft:

$M_1 = \left(L \cdot \frac{D_{drum}}{2} : i_{gb}\right) \cdot \eta_{gb} \cdot 9{,}81$ (Nm)	$M_1 = \left(31\,350 \cdot \frac{1{,}2}{2} : 24{,}6\right) \cdot 0{,}94 \cdot 9{,}81 = 7051$ Nm

Drum diam. $D_{drum} = 1{,}2$ m
Gearbox reduction $i_{gb} = 24{,}6$
Efficiency gearbox $\eta_{gb} = 0{,}98^3$

4. Hoisting speed of the load:

v (m/min)	$v = 60$ m/min

5. Wire rope speed on the drum:

$v_d = 2 \cdot v$ (m/min) $= (2 \cdot v) : 60$ (m/sec)	$v_d = 2 : 60 = 120$ m/min $= 120 : 60 = 2$ m/sec

6. Nos. of rev/min of the drum:

$n_d = v_d : (\pi : D_d)$ (rev/min)	$n_d = 120 : (\pi \cdot 1{,}2) = 31{,}84$ rev/min

7. Nos. of rev/min of the motor:

$n_{motor} = n_{dr} \cdot i_{gb}$ (rev/min)	$n_m = 31{,}84 \cdot 24{,}6 = 783$ rev/min

8. Inertia moment on motorshaft from: motor(s); brake sheave(s) and gearbox:

$J_{rot} = J_m + J_b + J_{gb}$ (kg m^2)	$J_{rot} = 24 + 16 + 6 = 46$ kg m^2

9. Reduced inertia moment on the motorshaft from the weight of the spreader plus load:

$J_L = (L \cdot v_d^2 \cdot \eta_{gb}) : \omega_1^2$ (kg m^2)	$J_L = (31\,350 \cdot 2^2 \cdot 0{,}94) : \left(\frac{783{,}2\pi}{60}\right)^2$
	$J_L = 117\,876 : 81{,}95^2 = 17{,}55$ kg m^2

10. $J_{\text{total}} = J_{\text{rot}} + J_{\text{L}}$ (kg m^2) $\qquad$ $J_{\text{tot}} = 46 + 17{,}55 = 63{,}55$ kg m^2

11. Braking is immediately started with the electric current, delivering the nominal motor torque. (The 2 motors deliver in total
$N = 720$ kW at
$n = 783$ rev/min)

$$M_{\text{nom}} = \frac{N \cdot 9550}{n} \text{ (Nm)} \qquad M_{\text{nom}} = \frac{720 \cdot 9550}{783}$$

$$\mathrm{M}_{\text{nom}} = M_{\text{el.braking torque}} \qquad M_{\text{nom}} = M_{\text{eb}} = 8781 \text{ Nm}$$

12. $\omega_{\text{mot}} = \dfrac{n}{60} \cdot 2\pi$ (rad/sec) $\qquad$ $\omega_{\text{mot}} = \dfrac{783}{60} \cdot 2\pi$ (rad/sec) $= 81{,}95$ rad/sec

13. The effective braking time is:

$$t_{\text{brake}} = \frac{\omega_{\text{mot}} \cdot J_{\text{total}}}{M_{\text{eb}} + M_1} \text{ (sec)} \qquad t_{\text{br}} = \frac{81{,}95 \cdot 63{,}55}{8781 + 7051} \text{ sec} = 0{,}329 \text{ sec}$$

14. Wire rope displacement on the drum during braking:

$$S_{\text{d}} = \tfrac{1}{2} \cdot v_{\text{d}} \cdot t_{\text{br}} \text{ (m)} \qquad S_{\text{d}} = \tfrac{1}{2} \cdot 2 \cdot 0{,}329 = 0{,}329 \text{ m}$$

15. Total displacement of spreader and load during electric braking in hoisting direction:

$$S_{\text{spr + L}} = S_{\text{d}} : 2 \text{ (m)} \qquad S_{\text{spr + L}} = 0{,}329 : 2 = 0{,}165 \text{ m}$$

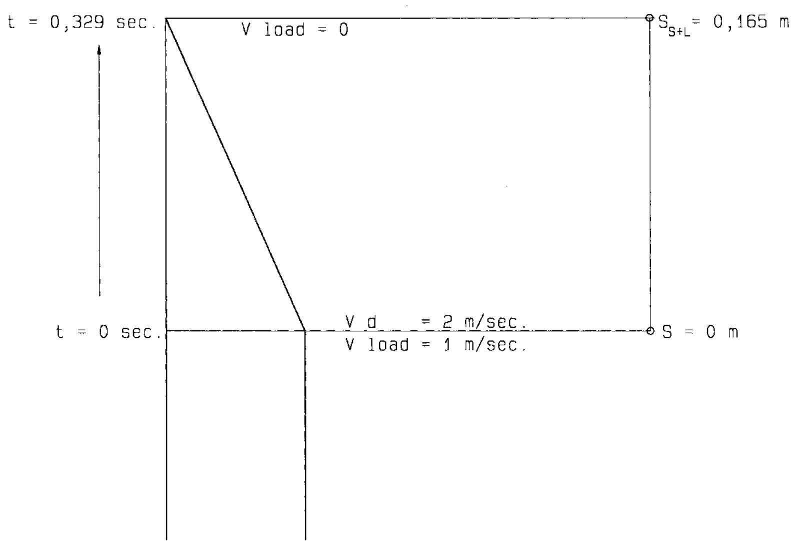


Fig. 4.4.2 Hoisting: electrical braking by full motor torque

4.5 Hoisting brakes

Hoisting the load; emergency stop

This is not as dangerous as in the lowering situation. The crane is hoisting the load and something occurs so that the crane driver must use the emergency push-button. Again, the hoisting mechanism does not brake electrically, it is an emergency stop. The load is at first decelerated by gravity, during the short time that is needed to activate the brake. The activated brake starts braking the load, but is starting from a lower speed than the normal hoisting speed. Assuming again that it is a container crane, with the reeving system shown in Fig. 4.5.1 (schematic).

Example

1. Weight of spreader plus load:
   Q (kg) $Q = 66\,000$ kg
2. Force on the wire ropes on the drum:

   L (kg) (see Fig. 4.5.1)

   $L = (Q:2) \cdot \eta_s$ (kg) $L = (66\,000:2) \cdot 0{,}95$
   $= 31\,350$ kg

   $\eta_s = \eta_{\text{sheaves + drum}} = n \cdot 0{,}99$ $(n = 5)$

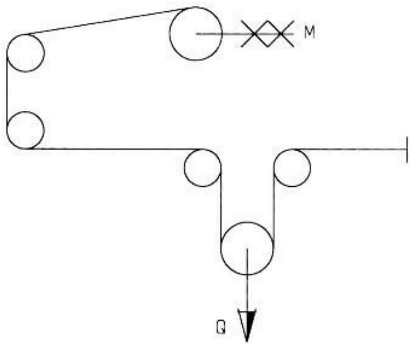


Fig. 4.5.1

3. Torque on motorshaft:

$$M_1 = \left(L \cdot \frac{D_{\text{drum}}}{2} : i_{\text{gb}}\right) : \eta_{\text{gb}} \cdot 9{,}81 \text{ (Nm)} \qquad M_1 = \left(31\,350 \cdot \frac{1{,}2}{2} : 24{,}6\right) \cdot 0{,}94 \cdot 9{,}81 = 7051 \text{ Nm}$$

Drum diam. $D_{\text{drum}} = 1{,}2$ m

Gearbox reduction $i_{\text{gb}} = 24{,}6$

Efficiency gearbox $\eta_{\text{gb}} = 0{,}98^3$

4. Hoisting speed of the load: v (m/min) $\qquad v = 60$ m/min

5. Wire rope speed on the drum:

$v_d = 2 \cdot v$ (m/min) $\qquad v_d = 2 \cdot 60 = 120$ m/min

$= (2 \cdot v) : 60$ (m/sec) $\qquad = 120 : 60 = 2$ m/sec

6. Nos. of rev/min of the drum:

$n_d = v_d : (\pi \cdot D_d)$ (rev/min) $\qquad n_d = 120 : (\pi \cdot 1{,}2) = 31{,}84$ rev/min

7. Nos. of rev/min of the motor:

$n_{\text{motor}} = n_d \cdot i_{\text{gb}}$ $\qquad n_m = 31{,}84 \cdot 24{,}6 = 783$ rev/min

8. Inertia moment on the motorshaft from: motor(s); brake sheave(s) and gearbox:

   $J_{rot} = J_m + J_b + J_{gb}$ (kg m^2) $\qquad$ $J_{rot} = 24 + 16 + 6$

   $= 46$ kg m^2

9. Reduced inertia moment on the motorshaft from the weight of the spreader plus load:

   $J_L = (L \cdot v_d^2 \cdot \eta_{gb}) : \omega_1^2$ (kg m^2) $\qquad$ $J_L = (31\,350 \cdot 2^2 \cdot 0{,}94)$:

   $$\left(\frac{783 \cdot 2 \cdot \pi}{60}\right)^2$$

   $J_L = 117\,876 : 81{,}95^2$

   $= 17{,}55$ kg m^2

10. $J_{total} = J_{rot} + J_L$ (kg m^2) $\qquad$ $J_{tot} = 46 + 17{,}55 = 63{,}55$ kg m^2

11. After pushing the emergency button, the load is decelerated by M_1 during Δt sec (activating time for the brake) with $\Delta\omega_2$ (rad/sec):

   $$\Delta\omega_2 = \frac{\Delta t \cdot M_1}{J_{rot}} \text{(rad/sec)} \qquad \Delta\omega_2 = \frac{0{,}3 \cdot 7051}{46}$$

   $= 46$ rad/sec

   $\Delta\omega_2$ is here negative $\qquad$ (negative)

12. The activated brake starts mechanical braking after Δt sec with a rotational speed on the motorshaft of:

   $\omega_3 = \omega_1 - \omega_2$ (rad/sec)— $\qquad$ $\omega_3 = \left(\dfrac{783 \cdot 2\pi}{60}\right) - 46$

   $\omega_1 = (n_m : 60) \cdot 2\pi$ (rad/sec) $\qquad$ $= 81{,}95 - 46$

   $= 35{,}95$ rad/sec

13. Nos. of rev/min of the motor- and brake-shaft is then:

$$n_2 = \omega_3 \cdot \frac{60}{2\pi} \text{ (rev/min)} \qquad n_2 = 36 \cdot \frac{60}{2\pi} = 344 \text{ (rev/min)}$$

14. The wire rope speed on the drum is then:

$$v_{d2} = \frac{n_2}{n_m} \cdot v_d \text{ (m/sec)} \qquad v_{d2} = \frac{344}{783} \cdot 2 = 0{,}88 \text{ m/sec}$$

15. The effective brake moment is:

$$M_{be} = \eta_b \cdot M_b \text{ (Nm)} \qquad M_{be} = 0{,}95 \cdot 19\,000 = 18\,050 \text{ Nm}$$

16. The effective braking time is:

$$t_{brake} = \frac{\omega_3 \cdot J_{total}}{M_{be} + M_1} \text{ (sec)} \qquad t_{br} = \frac{36 \cdot 63{,}55}{18\,050 + 7051} = 0{,}091 \text{ sec}$$

17. Total braking time:

$$t = (\Delta t + t_{br}) \text{ (sec)} \qquad t = 0{,}3 + 0{,}091 = 0{,}391 \text{ sec}$$

18. Wire rope displacement on the drum, during braking:

$$S_d = \Delta S_1 + \Delta S_2 \text{ (m)}$$

ΔS_1 = displacement on the drum during Δt (m)

ΔS_2 = displacement on the drum during deceleration while braking in t_{br} sec

$$\Delta S_1 = \frac{v_d + v_{d2}}{2} \cdot \Delta t \text{ (m)} \qquad \Delta S = \frac{2 + 0{,}88}{2} \cdot 0{,}3 = 0{,}432 \text{ m}$$

$$\Delta S_2 = \frac{v_{d2}}{2} \cdot t_{br} \text{ (m)} \qquad \Delta S_2 = \frac{0{,}88}{2} \cdot 0{,}091 = 0{,}04 \text{ m}$$

$$S_d = 0{,}432 + 0{,}04 = 0{,}472 \text{ m}$$

19. Total displacement of the spreader and load during emergency stop in hoisting direction (see Fig. 4.5.2):

$S_{spr+L} = S_d : 2$ (m) $\qquad S_{spr+L} = 0{,}472 : 2 = 0{,}236$ m

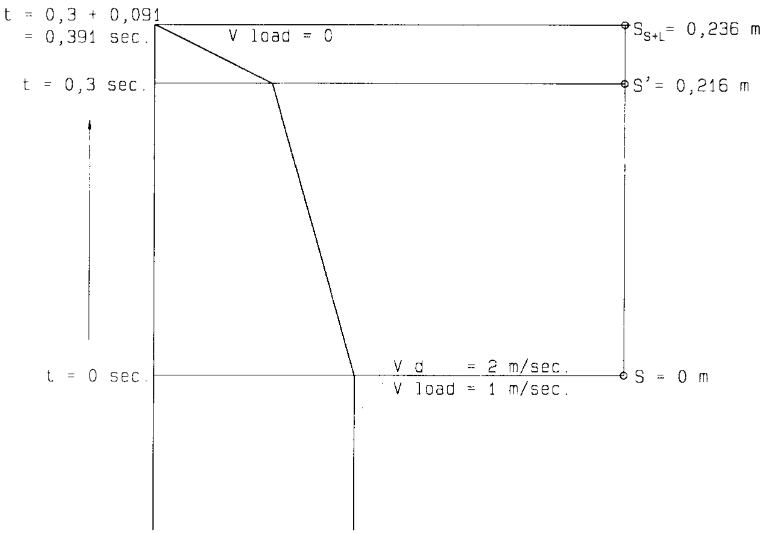


Fig. 4.5.2 Hoisting: emergency stop

4.6 Svendborg brakes

Svendborg brakes of Denmark manufacture disc brakes of a special construction which results in an activating time of $\Delta t = 0{,}1$ sec instead of $\Delta t = 0{,}3$ to 0,5 sec, when thrustor-activated disc brakes are used.

From Section 4.2, Lowering the Load; Emergency Stop (example point 11), we come to the following calculation:

	With $\Delta t = 0{,}3$ sec	With $\Delta t = 0{,}1$ sec
11. After pushing the emergency button, the load is accelerated by M_1 during Δt sec (activating time for the brake) with $\Delta\omega_2$ (rad/sec):		
$\Delta\omega_2 = \dfrac{\Delta t \cdot M_1}{J_{rot}}$ (rad/sec)	$\Delta\omega_2 = \dfrac{(0{,}3 \cdot 7051)}{46}$ $= 46$ rad/sec	$\Delta\omega_2 = \dfrac{(0{,}1 \cdot 7051)}{46}$ $= 15{,}3$ rad/sec

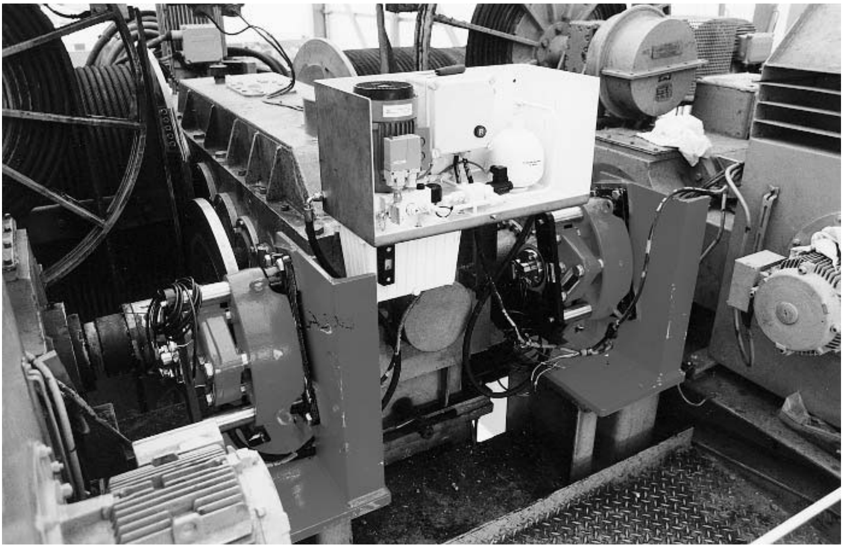


Fig. 4.6.1 Svendborg brakes

12. The activated brake starts mechanical braking after Δt sec with a rotational speed on the motorshaft of:

$$\omega_3 = (\omega_1 + \omega_2)\ (\text{rad/sec})$$
$$\omega_1 = (n_m : 60) \cdot 2\pi\ (\text{rad/sec})$$

$$\omega_3 = \left(\frac{783 \cdot 2\pi}{60}\right) + 46 = 81{,}95 + 46 = 127{,}95\ \text{rad/sec}$$

$$\omega_3 = \left(\frac{783 \cdot 2\pi}{60}\right) + 15{,}3 = 81{,}95 + 15{,}3 = 97{,}25\ \text{rad/sec}$$

13. The numbers of rev/min of the motor- and brake-shaft is then:

$$n_2 = \omega_3 \cdot \frac{60}{2\pi}\ (\text{rev/min}) \qquad n_2 = 128 \cdot \frac{60}{2\pi} = 1223\ \text{rev/min} \qquad n_2 = 97{,}25 \cdot \frac{60}{2\pi} = 929\ \text{rev/min}$$

14. The wire rope speed on the drum is then:

$$v_{d2} = \frac{n_2}{n_m} \cdot v_d\ (\text{m/sec}) \qquad v_{d2} = \frac{1223}{783} \cdot 2 = 3{,}123\ \text{m/sec} \qquad v_{d2} = \frac{929}{783} \cdot 2 = 2{,}37\ \text{m/sec}$$

15. The effective brake moment is:

$$M_{be} = \eta_b \cdot M_b\ (\text{Nm}) \qquad M_{be} = 0{,}95 \cdot 19\,000 = 18\,050\ \text{Nm} \qquad M_{be} = 0{,}95 \cdot 19\,000 = 18\,050\ \text{Nm}$$

16. The effective braking time is:

$t_{brake} = \dfrac{\omega_3 \cdot J_{total}}{M_{be} - M_1}$ (sec) $\quad t_{br} = \dfrac{128 \cdot 63{,}55}{18\,050 - 7051} = 0{,}740$ sec $\quad t_{br} = \dfrac{97{,}25 \cdot 63{,}55}{18\,050 - 7051} = 0{,}56$ sec

17. Total braking time:

$t = (\Delta t + t_b)$ (sec) $\quad t = 0{,}3 + 0{,}74 = 1{,}04$ sec $\quad t = 0{,}1 + 0{,}56 = 0{,}66$ sec

18. Wire rope displacement on the drum during braking:

$S_d = \Delta S_1 + \Delta S_2$ (m)

ΔS_1 = displacement on the drum during Δt (m)

ΔS_2 = displacement on the drum during deceleration whilst braking in t_{br} sec

$\Delta S_1 = \dfrac{v_d + v_{d2}}{2} \cdot \Delta t$ (m) $\quad \Delta S_1 = \dfrac{2 + 3{,}123}{2} \cdot 0{,}3 = 0{,}768$ m $\quad \Delta S_1 = \dfrac{2 + 2{,}37}{2} \cdot 0{,}1 = 0{,}22$ m

$\Delta S_2 = \dfrac{v_{d2}}{2} \cdot t_{br}$ (m) $\quad \Delta S_2 = \dfrac{3{,}123}{2} \cdot 0{,}74 = 1{,}155$ m $\quad \Delta S_2 = \dfrac{2{,}37}{2} \cdot 0{,}56 = 0{,}66$ m

$S_d = 0{,}768 + 1{,}155 = 1{,}923$ m $\quad S_d = 0{,}22 + 0{,}66 = 0{,}88$ m

19. Total displacement of spreader and load during emergency stop: in lowering direction (see Fig. 4.2.2)

$S_{spr+L} = S_d : 2$ (m) $\quad S_{spr+L} = 1{,}923 : 2 = 0{,}96$ m $\quad S_{spr+L} = 0{,}88 : 2 = 0{,}44$ m

4.7 Calculating the brake time and braking distance of a crane

Take a stacking crane running at high speed as an example.

Main characteristics

Weight of crane (t)		$W_1 = 650$ t
Weight of the load (t)		$W_2 = 40$ t
Total weight (t)		$W_t = 690$ t
Crane travelling speed	(m/min)	($v = 140$ m/min)
	(m/sec)	$v = 2{,}33$ m/sec
Crane travel resistance (kN/t)		$f = 5$ kg/t $= 0{,}05$ kN/t (take during braking $f = 3$ kg/t $= 0{,}03$ kN/t)

Fig. 4.7.1 Stacking crane on a rail terminal

Total efficiency of the gearings (η)	Let $\eta = 1$ in this case
Driving force of the wind: (kN)	
$F_w = (A \cdot c \cdot \eta) \cdot q$	
$q = 200\ \mathrm{N/m^2}$ ($v_w = 17{,}88$ m/sec)	$W = 16{,}90\ \mathrm{t} = 169\ \mathrm{kN}$
Wheel radius (R_w)	$R_w = 0{,}45$ m
Wheel diameter (D_w)	$D_w = 0{,}9$ m
Nos. of rev/min of the motors (n)	$n = 1800$ rev/min
Reduction between motor and wheel (i)	$i = \dfrac{n \cdot \pi \cdot D_w}{v}$
	$i = \dfrac{1800 \cdot \pi \cdot 0{,}9}{140}$
	$i = 36{,}33$
Rotating masses (J)	$J = 16\ \mathrm{kg\,m^2}$
Braking torque (M_b)	$M_b = 5000$ Nm

1. The travelling resistance during braking is:

   $F_1 = W_t \cdot f = 690 \cdot 30 = 20\,700\ \mathrm{N}$

   Reduced on the motorshaft this means:

   $$M_1 = F_1 \cdot R_w \cdot \frac{1}{i} = 20\,700 \cdot 0{,}45 \cdot \frac{1}{36{,}33} = 256{,}4\ \mathrm{Nm}$$

2. The 16 brakes deliver $M_b = 5000$ Nm as braking torque on the motorshafts.

3. The wind drives the crane with $W = 16{,}90$ t.

$$M_w = \frac{W \cdot R_w}{i} = \frac{169\,000 \cdot 0{,}45}{36{,}33} = 2093 \text{ Nm}$$

4. The inertia of the linear moving masses and of the rotating masses try to keep the crane moving during the deceleration time t. The influence of the linear moving masses is:

$$F_{\text{lin}} = m \cdot a = m \cdot \frac{v}{t}$$

$$F_{\text{lin}} = \frac{6\,900\,000}{9{,}81} \cdot \frac{2{,}33}{t}$$

$$F_{\text{lin}} = \frac{1\,638\,838}{t} \text{ N ('on the rails')}$$

Reducing to the motorshafts this is: $M_{\text{lin}} = F_{\text{lin}} \cdot R_w \cdot \frac{1}{i}$ (Nm)

$$M_{\text{lin}} = \frac{1\,638\,838}{t} \cdot 0{,}45 \cdot \frac{1}{36{,}33} = \frac{20\,300}{t} \text{ (Nm)}$$

5. The influence of the rotating masses is:

$$M_{\text{rot}} = \frac{J \cdot \omega}{t} = \frac{J \cdot \pi \cdot n}{30 \cdot t} = \frac{J \cdot n}{9{,}55 \cdot t} \text{ (Nm)}$$

$$M_{\text{rot}} = \frac{16 \cdot 1800}{9{,}55 \cdot t} = \frac{3015{,}7}{t} \text{ (Nm)}$$

6. The brakes and travelling resistance do retard the crane; the wind and the linear moving and rotating masses drive the crane.

$$M_1 + M_b = M_w + M_{\text{lin}} + M_{\text{rot}}$$

$$256{,}4 + 5000 = 2093 + \frac{20\,300}{t} + \frac{3015{,}7}{t}$$

$$3163{,}4 = \frac{23\,315{,}7}{t}$$

7. The crane will stop in approximately $t = 7{,}4\,\text{sec}$.
8. The braking distance after the brakes have come into action is:

$$S_t = v_0 \cdot t - \tfrac{1}{2}at^2 = v_0 t - \tfrac{1}{2} \cdot \frac{v}{t} \cdot t^2$$

$$S_t = \tfrac{1}{2} v \cdot t$$

$$S_t = \tfrac{1}{2} \cdot 2{,}33 \cdot 7{,}4 = 8{,}6\,\text{m}.$$

(A similar calculation can be carried out for the braking distance of a fast running trolley.)

Fig. 4.7.2 Programming an electric installation

4.8 The acceleration of a crane by wind at the beginning of an emergency stop

Assume that an emergency stop is necessary. A strong wind drives the crane; the crane driver hits the emergency push-button when the crane is running at nominal speed. The brakes come into full action after 0,3 sec. What will the crane travel speed be when the brakes come into full action?

Main characteristics

Example

Weight of crane (t)		$W_1 = 1300\,\text{t}$
Weight of load (t)		$W_2 = 40\,\text{t}$
Total weight (t)		$W_t = 1340\,\text{t}$
Crane travelling speed	(m/min)	($v = 45\,\text{m/min}$)
	(m/sec)	$v = 0{,}75\,\text{m/sec}$

Crane travelling resistance (kN/t) — $f = 5$ kg/t $= 0{,}05$ kN/t (take during braking) $f = 3$ kg/t $= 0{,}03$ kN/t

Total efficiency of the gearings (η) — Let $\eta = 1$, in this case

Driving force of the wind: (kN)

$F_w = (A \cdot C \cdot \eta) \cdot q$

$q = 275$ N/m^2 (W) — $W = 51$ t $= 510$ kN

($v_w = 21$ m/sec)

Wheel radius (R_w) — $R_w = 0{,}45$ m

Wheel diameter (D_w) — $D_w = 0{,}9$ m

Reduction between motor and wheel (i)

$$i = \frac{n \cdot \pi \cdot D_w}{v}$$

$$i = \frac{1800 \cdot \pi \cdot 0{,}9}{45}$$

$$i = 113$$

$J = \Sigma_{mom}$ of inertia of the rotating masses of motors, brake sheaves, couplings, etc. (kg m^2) — $J = 12$ kg m^2

1. The travelling resistance is:

$$F_1 = W_t \cdot f \text{ (kN)} \qquad F_1 = 1340 \cdot 0{,}03 = 40{,}2 \text{ kN}$$

2. Influence of the driving wind (kN) $W = 510$ kN

3. The influence of the linear moving masses is:

$$F_{lin} = m \cdot a = m \cdot \frac{v}{t} \text{ (N) (‘on the rails’)}$$

$$F_{lin} = \frac{13\,400\,000}{9{,}81} \cdot \frac{0{,}75}{t} = \frac{1\,024\,464}{t} \text{ (N)}$$

Reduced to the motorshafts this is:

$$M_{lin} = F_{lin} \cdot R_w \cdot \frac{1}{i} \text{ (Nm)}$$

$$M_{lin} \frac{1\,024\,464}{t} \cdot 0{,}45 \cdot \frac{1}{113} = \frac{4097{,}7}{t} \text{ (Nm)}$$

4. The influence of the rotating masses is:

$$M_{\text{rot}} = \frac{J \cdot \omega}{t} = \frac{J \cdot \pi \cdot n}{30t} = \frac{J \cdot n}{9{,}55t}$$

$$M_{\text{rot}} = \frac{12 \cdot 1800}{9{,}55 \cdot t} = \frac{2261{,}8}{t}$$

5. $$M_{\text{lin}} + M_{\text{rot}} = \frac{4079{,}7}{t} + \frac{2261{,}8}{t} = \frac{6341{,}5}{t} = 1{,}55 \cdot M_{\text{lin}}$$

   The influence of the rotating masses is in this case 55 percent of the linear moving masses.

6. Driving = retarding

   $F_{\text{w}} - F_1 = \text{m.a.}$

$$(510 - 40{,}2) = \frac{13\,400 \cdot 1{,}55}{9{,}81} \cdot a$$

$$a = \frac{9{,}81 \cdot (510 - 40{,}2)}{13\,400 \cdot 1{,}55} = 0{,}22\,\text{m/sec}^2$$

7. Over the period of 0,3 sec that the brakes need to come into action, the crane is accelerated by the wind with an acceleration of $a = 0{,}22\,\text{m/sec}^2$.

8. At the moment that the brakes come into action during this emergency stop, the crane travelling speed will be:

   $v_{\text{t}} = v_0 + a \cdot t$ (m/sec)

   $v_{\text{t}} = 0{,}75 + 0{,}22 \cdot 0{,}3 = 0{,}816\,\text{m/sec}$

9. The increase of the speed of the crane during the 0,3 sec is already

$$\Delta v = \frac{0{,}816 - 0{,}75}{0{,}75} \cdot 100 \text{ percent} = 8{,}8 \text{ percent.}$$

4.9 Storm pins and storm brakes

Section 3.6 shows how the power of the crane travelling motors should be calculated. In Section 4.7 the calculation for the braking distance of a crane was demonstrated. The influence of wind and storm can be calculated, using the information in Section 1.5. Referring back to Section 3.6 and resuming:

Weight of crane $W_1 = 1300$ t (unloaded)

Influence of side wind $F = (A \cdot C \cdot \eta) \cdot q$

$q = 275\ \mathrm{N/m^2}$ $F_w = 510$ kN

– Under storm conditions, windforce: 11

$q = 583\ \mathrm{N/m^2}$

(windspeed 110 km/h) $F_S = \frac{583}{275} \cdot 510 = 1080$ kN

Available motorpower: $\Sigma N = 24 \cdot 17 = 408$ kW
When platebrakes are built-in in the motors, or when open blockbrakes are installed, the nominal breaking torque is normally taken as: $M_b = 1{,}8 \cdot M_{motor}$

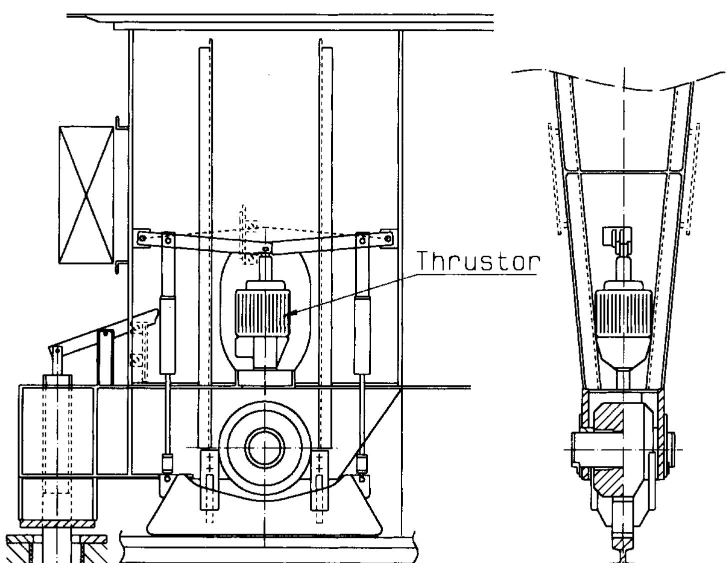


Fig. 4.9.1 Storm pin (left) combined with stormbrake of the brake-shoe type

Without taking the efficiency of the gearboxes and the resistance of the crane (3 kg /t) into account the braking force through the driven wheels onto the crane track is:

$$F = \frac{1{,}8 \cdot N}{v} = \frac{1{,}8 \cdot 408}{0{,}75} = 979\ \text{kN}$$

$$v = 45\ \text{m/min} = 0{,}75\ \text{m/sec}$$

However, the crane tends to topple over due to this very heavy storm; giving far more lower wheel loads on the leeside than on the windward side. This means in fact that less braking force is available on the rails, due to the decreased wheel loads on the windward side and – through that – a decreased adhesion coefficient between wheel and rail. So: extra storm brakes are needed. Take on landside and seaside one storm brake, each for at least $F = \frac{1}{4} \cdot F_S$ (kN).

Different types of storm brakes

Many types of storm brakes are available; among others there are:

1 The vertical pin type storm brake or stormpin

A vertical pin is put into an armoured pinhole next to the crane track. Normally this is done by hand. Vertical stormpins give an absolutely

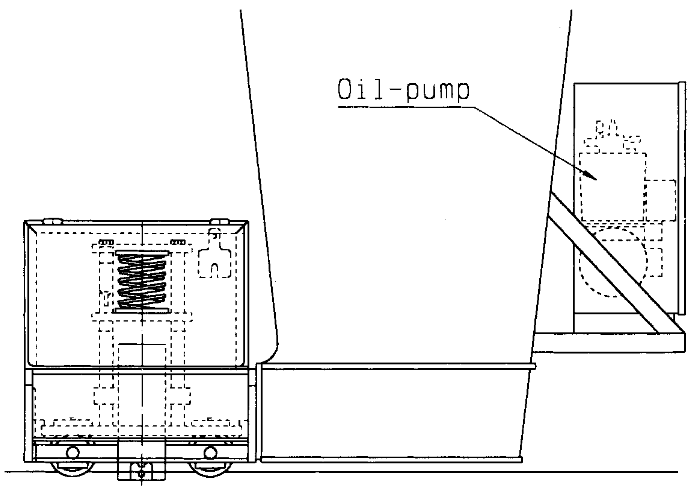


Fig. 4.9.2 Stormbrake of the rail clamp type

Fig. 4.9.3 Bubenzer rail clamp

safe system to prevent a crane drifting away in a storm or gale, but this system has the disadvantage that the crane first has to be driven to the position where the stormpin can be dropped into the stormpot. This is the reinforced hole in the quay which is destined to take up the stormpin. These stormpots are normally located on a centre to centre distance of approximately 50 m. In the worst case the crane has then to travel some distance against the heavy wind toward the next stormpot which is free. For this purpose the crane travelling motors must be strong enough to cover at maximum motor torque the distance toward the next stormpot.

2 The rail clamp type

With this type, hardened claws are pressed by springs against the sides of the crane rail. Hydraulic cylinders or other active elements release the claws from the rail sides, against the pressure of the springs.

3 The brake-shoe type

Here, a sturdy roll is fixed under the sill beam, directly above a rail shoe which is covered on the underside with friction material and which has a curved upperpart. A thrustor can lower the rail shoe onto the rail; which is done when the crane is in the rest-position. If the strong wind drives the crane aside, the roll touches the curved upperpart of the rail shoe and presses the whole part of the crane weight that is resting on the roll onto the brake shoe, thus giving a very high braking force.

Stormbrakes of types 2 and 3 work automatically. Normally they are activated some seconds after the crane has been stopped by ‘electric braking’ and after the crane travelling brakes have come into action. In regions where typhoons can be expected, it is necessary to provide storm-tiedowns. With these tiedowns vertical forces can be taken up in order to prevent cranes toppling over.

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Chapter 5

Standards

5.1 CEN

The CEN (Comité Européen de Normalisation) European committee for standardization is now the group which is developing European standards, and these standards are becoming recognized world-wide. The standards on cranes, conveyors and material handling equipment are also reviewed and renewed. CEN standards will, in the future, replace all standards from DIN; BS; NEN; NBN; NF; etc. and the FEM rules for the design of Hoisting Appliances, etc. The members of CEN are the national standards bodies of Austria; Belgium; Denmark; Finland; France; Germany; Greece; Ireland; Italy; Luxembourg; the Netherlands; Norway; Portugal; Spain; Switzerland; the United Kingdom; the Czech Republic; Iceland and Sweden.

CEN standards are:

EN 12077-2: 1998	Cranes safety – Requirements for health and safety – Part 2: Limiting and indicating devices
EN 12644-1: 2001	Cranes – Information for use and testing – Part 1: Instructions
EN 12644-2: 2000	Cranes – Information for use and testing – Part 2: Marking
ENV 1993-6: 1999	Eurocode 3: Design of steel structures – Part 6: Crane supporting structures

Draft European standards:

prEN 12644-3	Cranes. Safety. Requirements for inspection and use. Part 3. Fitness for purpose
prEN 12999	Cranes. Safety. Loader cranes
prEN 13000	Cranes. Safety. Mobile cranes
prEN 13001-1	Crane safety. General design. Part 1. General principles and requirements
prEN 13001-2	Crane safety. General design. Part 2. Load effects
prEN 13135-1	Cranes. Safety. Design. Requirements for equipment. Part 1. Electrotechnical equipment
prEN 13135-2	Cranes. Equipment. Part 2. Non-electrotechnical equipment
prEN 13155	Cranes. Safety. Non-fixed load lifting attachments
prEN 13157	Cranes. Safety. Hand powered cranes
prEN 13557	Cranes. Controls and control stations
prEN 13586	Cranes. Access
prEN 13852-1	Cranes. Offshore cranes. Part 1. General purpose offshore cranes
prEN 14238	Cranes. Manually controlled load manipulating devices

5.2 FEM

The Federation Européen de la Manutention has published a number of well known standards for Cranes, etc.

In FEM 1.001; 3rd Edition, Revised 1998, 10.01, the following *Rules for the Design of Hoisting Appliances* have been published.

Booklet

1. Object and scope.
2. Classification and loading on structures and mechanisms.
3. Calculating the stresses in structures.

4. Checking for fatigue and choice of mechanism components.
5. Electrical equipment.
6. Stability and safety against movement by the wind.
7. Safety rules.
8. Testloads and tolerances.
9. Supplements and comments to booklets 1 to 8.

FEM Section 2 gives the rules for Continuous Handling and Section 5 the rules for Mobile Cranes. FEM standards are very popular and are respected and used world-wide. However, since the European countries decided that the CEN standards should be developed and that all other standards on Cranes, like those in DIN, BS, NEN and NBN should no longer be developed, the publication of the very useful FEM standards will cease.

As CEN has so far only published the first draft standards on Cranes, the FEM standards still hold sway. The FEM standards on wind can be found in Section 1.5 of this book. In Section 7.6 a summary of the calculations on strength and fatigue are given.

FEM has prepared some modifications in their standards, to introduce the new methods as described in the future CEN standards.

5.3 ISO

ISO (International Standard Organization) is well known in the world and has special standards for Cranes. ISO member bodies are:

Austria	Israel	Republic of South Africa
Belgium	Japan	Spain
Czech/Slovakia	Mexico	Sweden
Finland	The Netherlands	Switzerland
France	New Zealand	former Soviet Union
Germany	Norway	United Kingdom
India	Poland	USA
Ireland	Romania	

The International Organization for Standardization is a world-wide federation of national standards institutes.

Drafts of ISO standards are circulated to the member bodies for approval, before they are accepted as ISO standards.

Some of the ISO standards on Cranes are:

ISO 4301-1: 1986	Cranes and lifting appliances – Classification. Part 1: General

ISO 4301-2: 1985	Lifting appliances – Classification – Part 2: Mobile cranes
ISO 4301-3: 1993	Cranes – Classification – Part 3: Tower crane
ISO 4301-4: 1989	Cranes and related equipment – Classification – Part 4: Jib cranes
ISO 4301-5: 1991	Cranes – Classification – Part 5: Overhead travelling and portal bridge cranes
ISO 4302: 1981	Cranes – Wind load assessment
ISO 4304: 1987	Cranes other than mobile and floating cranes – General requirements for stability
ISO 4305: 1991	Mobile cranes – Determination of stability
ISO 4306-1: 1990	Cranes – Vocabulary – Part 1: General
ISO 4306-2: 1994	Cranes – Vocabulary – Part 2: Mobile cranes
ISO 4306-3: 1991	Cranes – Vocabulary – Part 3: Tower cranes
ISO 4310: 1981	Cranes – Test code and procedures
ISO 7296-1: 1991	Cranes – Graphic symbols – Part 1: General
ISO 7296-1: 1991/Amd 1: 1996	
ISO 7296-2: 1996	Cranes – Graphical symbols – Part 2: Mobile cranes
ISO 7752-2: 1985	Lifting appliances – Control – Layout and characteristics – Part 2: Basic arrangement and requirements for mobile cranes
ISO 7752-2: 1985/Add 1: 1986	
ISO 7752-3: 1993	Cranes – Control – Layout and characteristics – Part 3: Tower cranes
ISO 7752-4: 1989	Cranes – Controls – Layout and characteristics – Part 4: Jib cranes
ISO 7752-5: 1985	Lifting appliances – Controls – Layout and characteristics – Part 5: Overhead travelling cranes and portal bridge cranes

ISO 8087: 1985	Mobile cranes – Drum and sheave sizes
ISO 8306: 1985	Cranes – Overhead travelling cranes and portal bridge cranes – Tolerances for cranes and tracks
ISO 8566-1: 1992	Cranes – Cabins – Part 1: General
ISO 8566-2: 1995	Cranes – Cabins – Part 2: Mobile cranes
ISO 8566-3: 1992	Cranes – Cabins – Part 3: Tower cranes
ISO 8566-4: 1998	Cranes – Cabins – Part 4: Jib cranes
ISO 8566-5: 1992	Cranes – Cabins – Part 5: Overhead travelling and portal bridge cranes
ISO 8686-1: 1989	Cranes – Design principles for loads and load combinations – Part 1: General
ISO 8686-3: 1998	Cranes – Design principles for loads and load combinations – Part 3: Tower cranes
ISO 8686-5: 1992	Cranes – Design principles for loads and load combinations – Part 5: Overhead travelling and portal bridge cranes
ISO 9373: 1989	Cranes and related equipment – Accuracy requirements for measuring parameters during testing
ISO 9374-1: 1989	Cranes – Information to be provided – Part 1: General
ISO 9374-4: 1989	Cranes – Information to be provided – Part 4: Jib cranes
ISO 9374-5: 1991	Cranes – Information to be provided – Part 5: Overhead travelling cranes and portal bridge cranes
ISO 9926-1: 1990	Cranes – Training of drivers – Part 1: General
ISO 9927-1: 1994	Cranes – Inspections – Part 1: General
ISO 9928-1: 1990	Cranes – Crane driving manual – Part 1: General
ISO 9942-1: 1994	Cranes – Information labels – Part 1: General
ISO 9942-3: 1999	Cranes – Information labels – Part 3: Tower cranes

ISO 10245-1: 1994	Cranes – Limiting and indicating devices – Part 1: General
ISO 10245-2: 1994	Cranes – Limiting and indicating devices – Part 2: Mobile cranes
ISO 10245-3: 1999	Cranes – Limiting and indicating devices – Part 3: Tower cranes
ISO 10245-5: 1995	Cranes – Limiting and indicating devices – Part 5: Overhead travelling and portal bridge cranes
ISO 10972-1: 1998	Cranes – Requirements for mechanisms – Part 1: General
ISO 10973: 1995	Cranes – Spare parts manual
ISO 11630: 1997	Cranes – Measurement of wheel alignment
ISO 11660-1: 1999	Cranes – Access, guards and restraints – Part 1: General
ISO 11660-2: 1994	Cranes – Access, guards and restraints – Part 2: Mobile cranes
ISO 11660-3: 1999	Cranes – Access, guards and restraints – Part 3: Tower cranes
ISO 11660-5: 2001	Cranes – Access, guards and restraints – Part 5: Bridge and gantry cranes
ISO 11661: 1998	Mobile cranes – Presentation of rated capacity charts
ISO 11662: 1995	Mobile cranes – Experimental determination of crane performance – Part 1: Tipping loads and radii
ISO 11994: 1997	Cranes – Availability – Vocabulary
ISO 12210-1: 1998	Cranes – Anchoring devices for in-service and out-of-service conditions – Part 1: General
ISO 12210-4: 1998	Cranes – Anchoring devices for in-service and out-of-service conditions – Part 4: Jib cranes
ISO 12210-4: 1998/Cor 1: 2000	
ISO 12478-1: 1997	Cranes – Maintenance manual – Part 1: General
ISO 12480-1: 1997	Cranes – Safe use – Part 1: General

ISO 12482-1: 1995	Cranes – Condition monitoring – Part 1: General
ISO 12485: 1998	Tower cranes – Stability requirements
ISO 13200: 1995	Cranes – Safety signs and hazard pictorials – General principles
ISO 15513: 2000	Cranes – Competency requirements for crane drivers (operators), slingers, signallers and assessors
ISO/TS 15696: 2000	Cranes – List of equivalent terms

5.4 DIN; BS; JIS

These standards are also important but are only mentioned here.

– DIN	Germany	Deutsche Industrie Normen
– BS	United Kingdom	British Standards
– JIS	Japan	Japanese Industry Standards

Furthermore, there are national standards on cranes in almost every country.

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Chapter 6

Sagging and Slapping of the Wire Ropes; Rock and Roll of the Spreader; Machinery Trolleys versus Wire Rope Trolleys; Twin-lift; Positioning; Automatic Equipment Identification (AEI)

6.1 Sagging and slapping of the wire ropes; other hoist wire rope systems for container quay cranes and grab unloaders

Section 2.1 showed one hoist wire rope system for container cranes and one for grab unloaders with a main- and auxiliary trolley. For container quay cranes further wire rope systems for the hoisting wire ropes are used.

Figure 6.1.1 gives a schematic diagram of a rather common hoist wire rope system in which the container is hanging on 12 falls instead of 8 falls as shown in Section 2.1. Figure 6.1.2 shows two auxiliary trolleys, which run at half the speed of the main trolley. These auxiliary trolleys are intended to decrease the sagging and slapping of the hoist and trolley wire ropes.

The higher the hoist and trolley speeds are, and the longer the trolley travel range is, the more the sagging and slapping of the wire ropes will influence the throughput of the crane. A very good system is shown in Fig. 6.1.3 with fully supported hoist- and trolley wire ropes, which gives the best possible protection against their sagging and slapping in all circumstances.

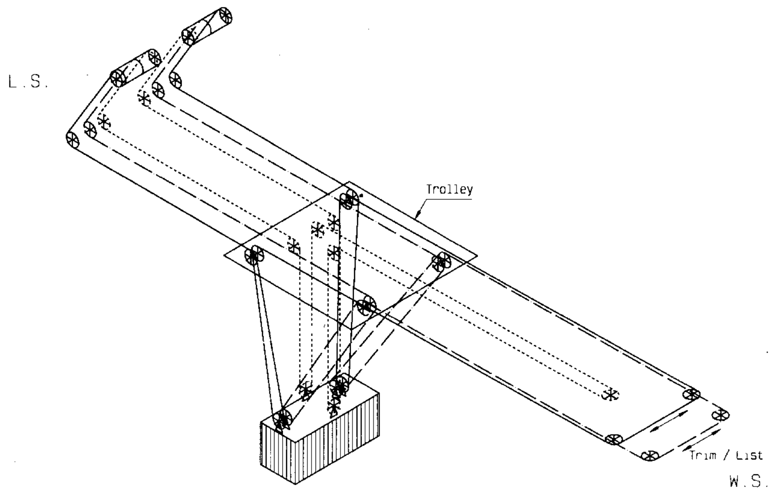

Fig. 6.1.1 Headblock hanging on 12 wire rope falls

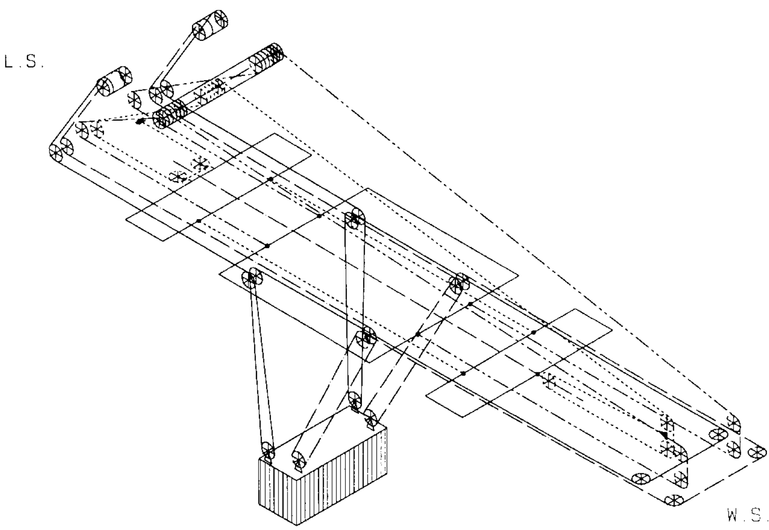

Fig. 6.1.2 Wire rope support with two catenary trolleys

Grab unloaders

The same parameters and considerations apply to grab unloaders as to container quay cranes. The weights of the trolleys of grab unloaders vary widely, they can be quite heavy.

Fig. 6.1.3 Fully supported wire ropes

When the hoisting machinery is installed on the trolley itself, it is possible to give the grab a cross-traversing. The unloader with main- and auxiliary trolley (see Fig. 6.1.4) has all the advantages of rope trolleys. Because of the presence of the auxiliary trolley the free hanging

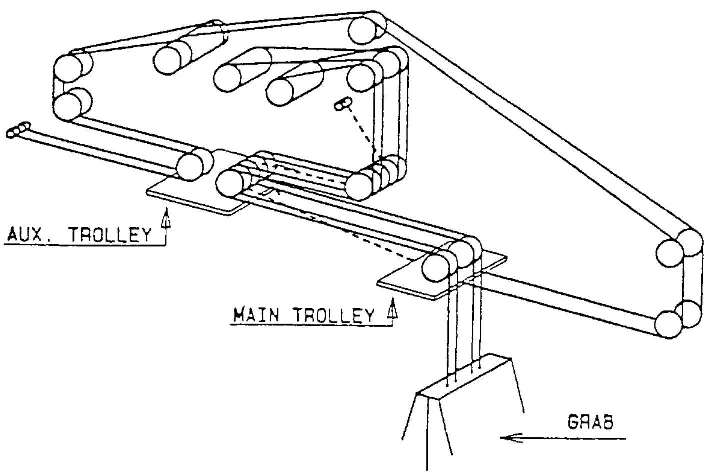

Fig. 6.1.4 Rope reeving system of a grab unloader

and slapping wire rope length is already limited. The system with two auxiliary trolleys for wire rope support, or a system with fully supported wire ropes is not used in grab unloaders.

Other wire rope systems which sometimes are used in grab unloaders are:

– the 'fleet through' reeving system
– the 'in bight of line' reeving system.

The 'fleet through' reeving system

'Fleet through' reeving systems are simple. However, because the close wire rope, and the hold wire rope, run through the sheaves of the grab when the trolley is travelling, this means that extra wear and tear is caused through a greater number of bendings. This particularly affects the close wire rope. In addition to the increased wear and tear, especially on the low diameter sheaves in the grab, there are further problems that occur when the close wire rope comes into contact with the transported material, for example, ore, coal, or other abrasive materials. The hold wire rope runs in this system over one sheave, which is fastened in or above the top of the grab.

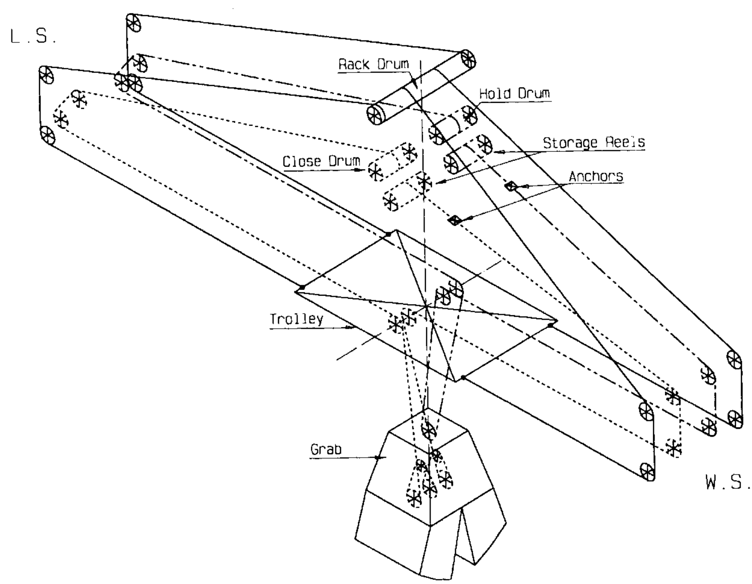

Fig. 6.1.5 'Fleet through' reeving system

Because of this extra wear on the ropes, motor driven storage reels are mounted in the boom, and after each 10 000 tons or so of transported material, the wire ropes are reeved through a specified length. This shifts the abraded wire rope along so that the same region of wire rope is not continuously abraded. A considerable amount of work is involved in this process, and all the ropes must be carefully measured and cut off at the same length.

'In bight of line' reeving system

The 'in bight of line' reeving system does not have the disadvantages of the 'fleet through' system, however here the close- and hold-drum have to be synchronized with the rack or trolley travelling drum. They have to move when the trolley is traversing, otherwise the grab will move vertically or downwards. Figure 6.1.6 illustrates the 'in bight of line' reeving system.

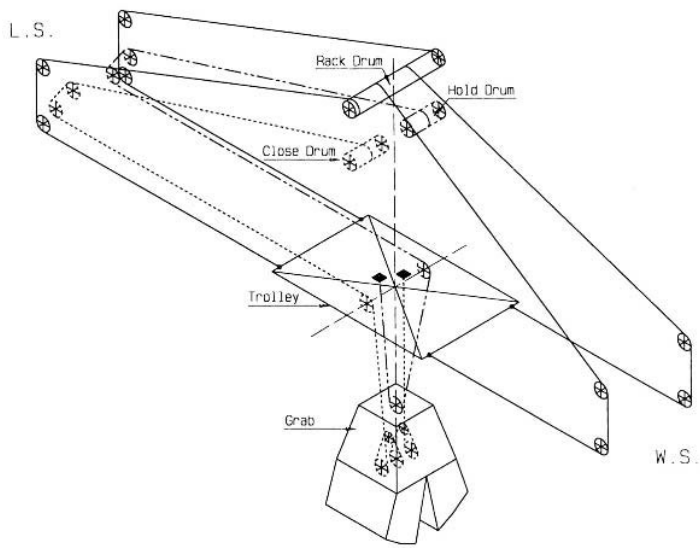

Fig. 6.1.6 'In bight of line' reeving system

6.2 The rock and roll of the spreader

In the wire rope reeving system for a normal container crane, the wire ropes, running down from the spreader towards the trolley, diverge somewhat, as shown in Fig. 6.2.1. When accelerating or decelerating

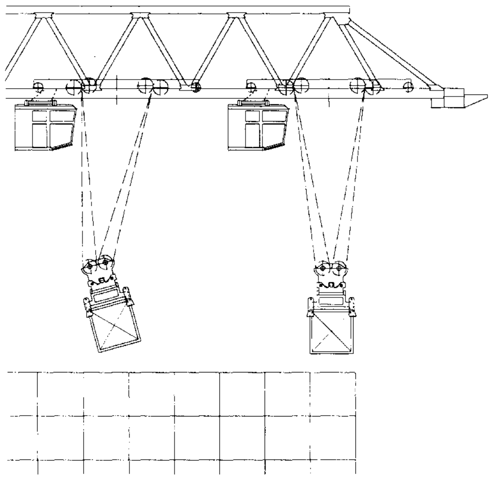

Fig. 6.2.1 The rock and roll of a spreader

the trolley, the spreader tends to swing and to rock. Because of the divergence of the wire ropes, the spreader tends to roll somewhat during accelerating and decelerating.

The greater the accelerating or decelerating and the trolley speeds are, the more hindrance will be experienced by the crane driver from the rocking and rolling of the spreader.

6.3 Advantages and disadvantages of machinery trolleys versus wire rope driven trolleys

The advantages or disadvantages of each system can be seen from a comparison between the systems.

Container quay cranes

	A Machinery trolley	B Semi-rope trolley	C Full-rope trolley
System	Complete hoisting and trolley travelling mechanism mounted on the trolley	Complete hoisting mechanism mounted in machinery house on the bridge girder(s) Trolley travelling by means of motors, driving the trolleywheels	Complete hoisting mechanism and trolley travelling mechanism mounted in machinery house on the bridge girder(s)

	A Machinery trolley	B Semi-rope trolley	C Full-rope trolley
Weight of trolley plus cabin (depending on speeds and hoist cap.)	Approx. 52–80 t	Approx. 26 t	Approx. 22–36 t
Trolleywheels	Driven	Driven	Non-driven
Wheelslip	Possible	Possible	Not possible
Slewing of containers	Possible	Not possible	Not possible
Max. trolley acceleration	Normal 0,5 m/sec	Normal 0,5 m/sec^2	Up to 1,2 m/sec^2
Greasing of trolley rail	Not possible	Not possible	Possible, giving less wear and tear of wheels and rails
Current supply to trolley	A heavy system with many flexible cables is necessary for all current supply plus control	Only current supply for trolley travelling, lighting, heating plus control is necessary	Only current supply for lighting, heating plus control
	If the trolley speed is above $v = 200$ m/min, some motor driven cable trolleys become necessary in the festoon system	If the trolley speed is above $v = 200$ m/min, it can become necessary to use some motor driven cable trolleys in the festoon system	If the trolley speed is above $v = 240$ m/min, it can be necessary to use some motor driven cable trolleys in the festoon system
Trolley travelling length	Unlimited	Limited through the eventual sagging and slapping of the wire ropes	Limited through the eventual sagging and slapping of the wire ropes
		However: Preventing sagging and slapping by 2 auxiliary trolleys or by full-supported wire ropes allows a far greater trolley travelling length	*However*: Preventing sagging and slapping by 2 auxiliary trolleys or by full-supported wire ropes allows a far greater trolley travelling length

Note: Semi-Machinery Trolley

It is also possible to install the complete hoisting mechanism on the trolley and to prevent the slip of the trolley wheels by using wire ropes. In the trolley travelling mechanism for driving the trolley, measures have then to be taken to prevent the sagging and slapping of these wire ropes. The trolley travelling mechanism comprising the motor(s), gearbox and wire rope drum, can be installed in the machinery house on the bridge. It can also be positioned on the trolley itself, which produces a heavier trolley but a simpler wire rope system. However the wire ropes are then not easy to support.

6.4 Container transport with twin-lift spreaders; long twin-lift; Bramma Tandemlift. Connecting the spreader to the headblock

Container vessels are equipped with 20 foot and 40 foot cells. As there are a large number of 20 foot containers to be transported, the stevedores started to stow two 20 foot containers into one 40 foot cell. The crane builders and spreader builders reacted by the employment of twinlift spreaders. These telescopic spreaders have twistlocks at both ends and retractable flippers. In the middle of the spreader a double set of retractable twistlocks is mounted. Handling one 20 foot container and one 40 foot container is done with the four twistlocks on the spreader-ends.

When two 20 foot containers have to be handled simultaneously, the telescopic spreader is interlocked on the twin-lift position and the double sets of retractable twistlocks in the middle of the spreader are lowered. Now the crane driver can handle two 20 foot containers simultaneously, giving a higher level of production, and higher throughput.

The flipper actuators must be oversized and very strong in order to achieve a high throughput. With twin-lift handling, the throughput of

Fig. 6.4.1 Twin-lift spreader

the container quay crane can be increased by some 15 percent. However, not every container crane can be used for twin lift! When handling one empty 20 foot container, plus one full but eccentrically loaded 20 foot container, weighing 25 tonnes, and a spreader plus headblock weighing approximately 10 tonnes, this produces an extremely large difference in the load on the hoisting ropes. With a single box main girder and boom, with a railgauge of approximately four metres and a low-weight trolley, it is possible to imagine the difficulties that can arise when the containers are eccentrically loaded. A wheel-driven trolley can have severe wheel slippage. Figure 6.4.2 shows this.

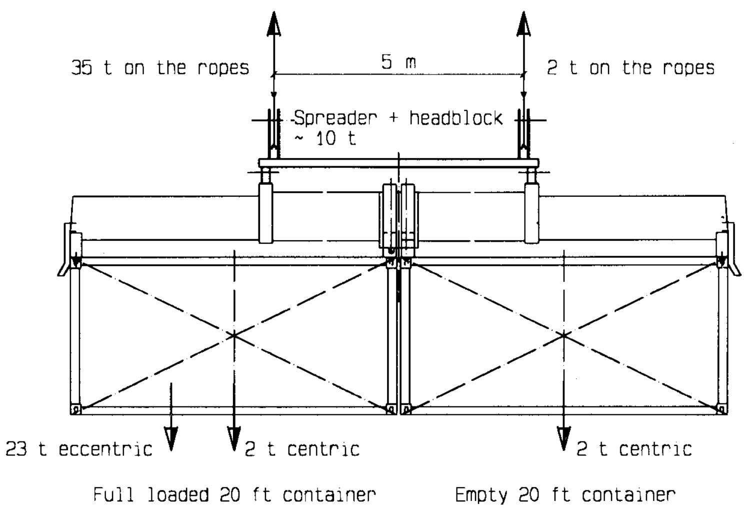

Fig. 6.4.2 Twin-lift: worse case

When handling a twin-lift spreader, a wide single box girder and boom, preferably 5,1 m railgauge, or a wide double box girder or double plated girder should be used. All users must be aware of the eccentric loading of containers. This eccentricity can be 10 percent of container length and width.

Also, as previously mentioned, 25 tonnes as the given weight for a twenty-foot container is no longer an accurate maximum. Often many containers weigh 30 tonnes rendering the twin-lift problems much worse than previously mentioned in Fig. 6.4.2. This can mean that the distance between the ropes should be more than five metres. The weight of a container in which liquids are packed can exceed 35 tonnes!

Long Twin-lift

The newest development in the twin-lift spreaders is the long twin-lift spreader, which has been fully patented by Stinis–Krimpen BV, Netherlands. With the 'long twin-lift' the two full-loaded 20 foot containers which are hanging underneath the spreader can be up to a distance of 1600 mm from each other. This can be done after having picked up the containers also in the air.

For vessels which have 20 foot container bays on deck separated for more efficient lashing, the Stinis long-twin-lift spreader can handle these two containers in one lift. It becomes easy to control the doors and seals of 20 foot containers with 'back to back' standing doors. The flipper actuators must again be very strong and oversized. This is necessary to achieve a high throughput.

Large guide rolls on the spreader are required to increase the handling speed. Automatic greasing/lubrication is important to reduce wear and tear as well as maintenance.

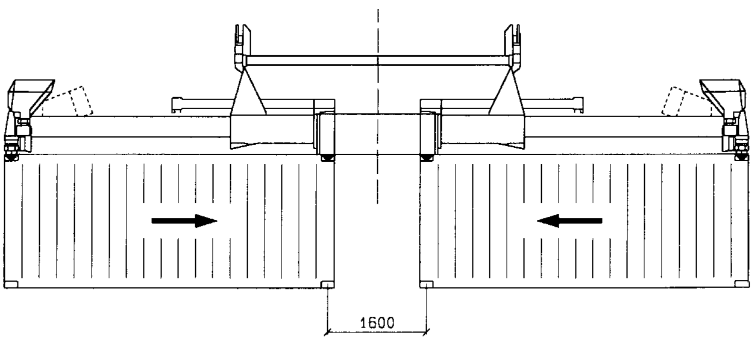

Fig. 6.4.3 Stinis Long Twin-lift spreader

Bromma Tandemlift

Bromma has introduced the Tandem line, a twin-lift spreader that can handle two 40 foot or 45 foot containers simultaneously, side-by-side. This spreader is designed to work on the deck – as well as on the containers in the cells.

The distance between the two side-by-side containers can be adjusted from 0 to 1200 mm and a 350 mm container height difference can be reached when picking up the containers or lowering the containers on trailers, AGVs, etc. The headblock of this rather heavy spreader has to be of a special design.

Fig. 6.4.4 Stinis Long Twin-lift spreader in action

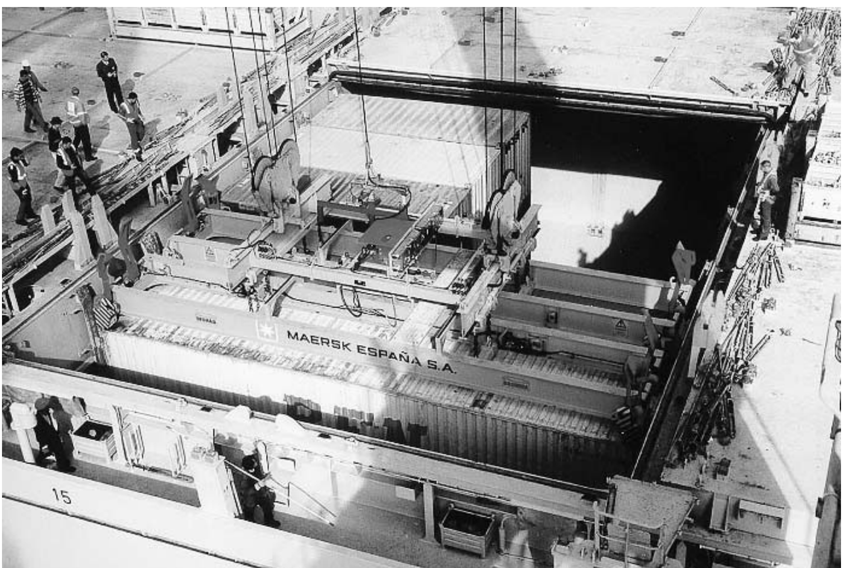

Fig. 6.4.5 Bromma Tandemlift

Fig. 6.4.6 Bromma Rackamatic

Fig. 6.4.7 Bromma telescopic spreader with grapple arms

Connecting the spreader to the headblock

The spreader can be connected to the headblock with horizontal pins which are protected by limit switches, or by four twistlocks which are also protected by limit switches. These twistlocks can be manually driven or driven by hydraulic cylinders which can be controlled by the crane driver. The spreader cable that comes down from the trolley has to be connected to the spreader by means of a plug and a receptacle.

These actions can also be automated. Bromma of Sweden has developed together with Kheops a fully patented automatic connector for a container crane; the Rackamatic. The upper section of the Rackamatic is connected to the headblock, the lower part, to the spreader. When the Rackamatic is used, the connection between headblock and spreader is by four twistlocks. These are controlled by the crane drive.

6.5 Sway and swing; automation of the trolley travelling mechanism

A load hanging on the wire ropes will sway due to wind, but also due to the acceleration and deceleration of the trolley. A grab always hangs on vertical wire ropes, normally four ropes are used. A spreader is hanging on eight, or more, wire ropes, which can hang vertically, but usually they diverge from the spreader towards the trolley. There, where the load (grab, spreader, or spreader with container) is hanging on vertical ropes the sway follows the rules of mathematical oscillation.

When handling eccentrically loaded containers or ‘twin-lifts’ with one heavy loaded and one light loaded or empty container, another feature arises – ‘swing’, during accelerating and decelerating of the trolley travel motion. This occurs particularly when the acceleration and deceleration is high, and the wire ropes are hanging vertically. This phenomenon can be very inconvenient.

It is usual for the trolley travelling mechanism to be automated in big grab unloaders that are used to unload from the holds of ships into rather wide bunkers or hoppers. For container ship-to-shore cranes this sort of automation is not yet routine as there are very narrow tolerances which have to be held when positioning on an AGV or trailer, which require the skill of an operator rather than an automatic system. However a number of manufacturers have developed systems for this type of automatic positioning.

Normally they work via camera(s) under the trolley and reflectors on the spreader. The deviation from the reflected light beams gives the trolley travel mechanism an indication as to how the acceleration/deceleration

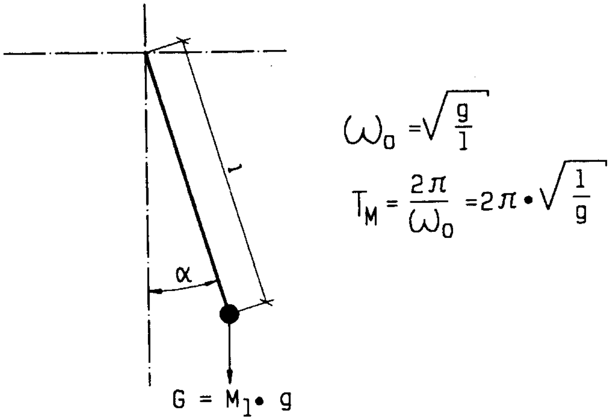

Fig. 6.5.1 Mathematical oscillation

Where: T_M = the oscillation time in sec. for the total oscillation ('to and from'). The oscillation time is a function of the pendulum length between the centre of rotation of the wire rope sheaves on the trolley and the centre of gravity of the load. If the wire ropes are diverging from the wire rope sheaves on the spreader, towards the wire rope sheaves on the trolley, the oscillation time will decrease, tending to cause less sway in the spreader and container.

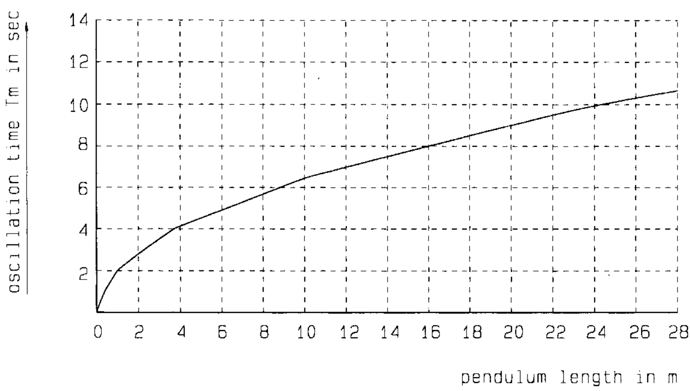

Fig. 6.5.2 Mathematical oscillation time

should be regulated to keep the spreader and container 'swayless' and/or 'swingless' and to stop it exactly in the correct position.

Note: The manufacturer Holec, Ridderkerk (now HMA), The Netherlands, as well as Dr Schichi Isomura; professor in the Department of

Fig. 6.5.3 Automatic trolley positioning: camera and spotlights under the trolley

Fig. 6.5.4 Automatic trolley positioning: reflector on the spreader

Mechanical Engineering; Takamatsu National College of Technology in Japan have issued the following document:

SWAY CONTROL

Sway control versus anti-sway

In order to move the load to the target position an accelerating force is needed. The only way to produce such a force is by developing a sway angle. Therefore, sway is a normal phenomenon in load handling which should be controlled instead of defeated.

Theory of sway

The model is easiest to understand by looking at the pendulum as a circular movement of mass m around the point of suspension on the trolley (with circular speed ω).

The forces working at m, perpendicular to the radius, give a momentum $T = F * l$ accelerating the movement. The inertia of the system is: $J = m * l^2$. Therefore, the angular acceleration becomes: $\mathrm{d}\omega/\mathrm{d}t = T/J = F/(\mathrm{m} * l) = a/l$.

There are four sources that can give an acceleration (perpendicular to the radius):

1. Gravity: $g * \sin\phi$
2. Acceleration of trolley: $a_k * \cos\phi$
3. Coriolis acceleration: $2 * v_1 * \omega$
4. Windforce: $F_w * \cos\phi/m$

v_1 is the velocity at which the load (mass m) is moving towards the midpoint of the circle (the hoist speed). At constant rope-length this third term will be zero.

$$\frac{\mathrm{d}\omega}{\mathrm{d}t} = \frac{-g\sin\phi - a_k\cos\phi + 2v_l\omega + \dfrac{F_w}{m}\cos\phi}{l}$$

The signs in this formula apply to the definitions below:

a_k: acceleration of the trolley in m/sec^2, positive when accelerating to the right-hand side;

ω: angular velocity in rad/sec, positive when rotating anti-clockwise (load moves to the right-hand side);

ϕ: angle in rad, positive when the load is at the right-hand side of the point of suspension on the trolley. Zero when the load is right underneath the trolley;

F_w: windforce acting on load in N, positive when the load would be moved to the right-hand side;

v_l: hoist speed in m/sec, positive when the load is moving toward the trolley;
l: radius in m;
g: gravitational acceleration in m/sec^2, positive when pointing downwards.

In the formula there is no term accounting for damping, however a sway will damp out spontaneously. According to the formula the *acceleration* of the trolley is the only independent factor that influences the sway.

Development of simulation model
From the equation a blockdiagram can be drawn. As the sway angles normally will be under 20 degrees the assumption can be made:

sin phi = phi (in radians); cos phi = 1

The windforce can be treated as a disturbance and not be included in the blockdiagram:

$$\frac{d\omega}{dt} = \frac{-g\phi - a_k + 2v_l\omega}{l}$$

Development of Holec sway control method
The first objective is to precalculate a route that brings the load at the target position in such a way that there is (theoretically) no residual sway. When the rope-length is constant the next simple solutions can be found. Because times and speeds are known, the elapsed distance can be easily calculated. The figures are drawn for acceleration to set speed only because deceleration from set speed will be symmetrical. Note that the figures are results of simulations which can be compared with the simulation results of MHI. One distinctive difference is the absence of a tail.

Varying rope-length
When the rope-length is varying some new phenomena arise. From the block diagram it can be seen that there is a big difference in behaviour between the situations of:

- constant rope-length *and* zero hoist speed
- varying rope-length *and* non-zero hoist speed.

When the hoist speed is non-zero the Coriolis acceleration will greatly influence the load!

Compensation for varying rope-length
When the rope-length is fixed all signals of trolley-distance, trolley-speed, sway angle, angular velocity and cycle time follow simple formula and can easily be predicted.

Because overlay between trolley travel and hoisting is required a solution must be found for dealing with varying rope-length. As noted before the acceleration of the trolley is the only independent factor that influences the sway.

The concept of compensation is based on the next equation:

$$\frac{d\omega}{dt} = \frac{-g\phi - a_{k0}}{l_0} = \frac{-g\phi - a_k + 2v_l\omega}{l}$$

The first term contains the values for a reference sway with a fixed length l_0. When the length is fixed the hoist speed v_l will be zero too. From this equation a_k can be solved in such a way that with varying length l the value of $d\omega/dt$ will stay equal to the value it would have with the fixed length l_0. When $d\omega/dt$ is equal at any moment, then the resulting ω and ϕ must be equal too:

$$a_k = \left(\frac{l}{l_0} - 1\right) g\phi + \frac{l}{l_0} a_{k0} + 2v_l\,\omega$$

where

$$\phi = \phi_{max}(1 - \cos(\omega_0 t)) \quad \text{with} \quad \phi_{max} = \frac{a_{k0}}{g}$$

$$\omega = \omega_{max}\sin(\omega_0 t) \quad \text{with} \quad \omega_{max} = \omega_0\phi_{max}$$

$$\omega_0 = \sqrt{\frac{g}{l_0}} \qquad T = 2\pi\sqrt{\frac{l_0}{g}}$$

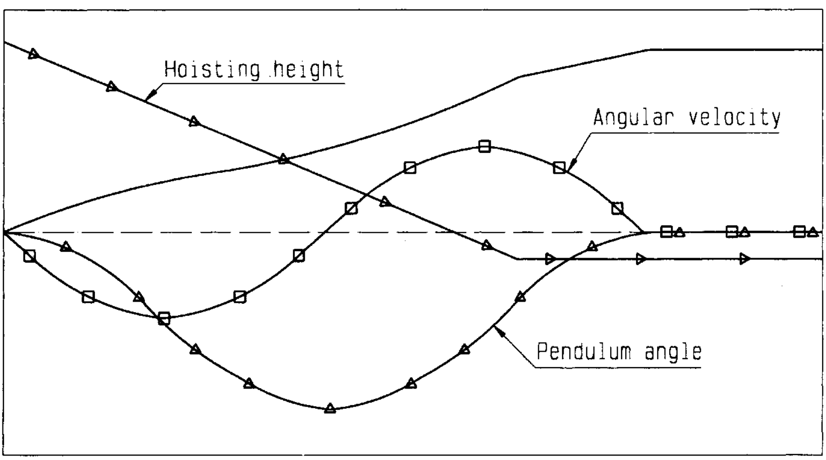

Fig. 6.5.5 Simulation long curve with variable rope-length

The integration of a_k to speed and position is possible. Although it will give rather complicated formulae, the calculation of these variables for a given value of time t will not be a problem for a computer.

Result of simulation
Employing the theory of the last paragraph, simulations have been carried out. It is found that when the hoist is moving only during a part of the trolley travel interval a correction must be deduced. For the simulations the fixed reference length of 10 m is used.

6.6 The positioning of a hoisting mechanism; automation

The hoisting height of the biggest container cranes can be as high as 47 m above the quay level and 21 m below quay level into the cells of a ship. This means that it becomes hard for the crane driver to judge the exact distance from the underside of the spreader, or the underside of the container hanging under the spreader to the AGV, or trailer on the quay, and the height of the container in the cell of the ship. Therefore it is useful for the operator to have a system in the hosting winch which measures the height and a display in his cabin that indicates this. Height measurement is executed by pulse counters or incremental encoders on the hoist motors or by laser under the trolley.

These systems are very accurate; and when used in conjunction with a metal finger underneath the spreader, they give a signal that the spreader has almost reached the container which has to be picked up. Furthermore 'Safe distances' can be built-in in the hoisting–lowering–trolley travelling range. By using these techniques the spreader will slow down if it reaches the AGVs and trailers and will stop if there is the danger that, for instance, a sillbeam will be hit.

Together with these positioning devices and built-in automation the crane driver can pick the container up as usual and start hoisting. After a certain distance a knob is pushed on the controller handle which switches in the automation. The automated system operates until the moment that the limit switch slows the hoist motor down and stops it, or to the moment that the crane driver takes the motion of the spreader back into manual control, from the automation. When lowering, the reversed pattern can be followed.

6.7 Automatic positioning for crane travelling mechanisms

Container terminals are increasingly using Automatic Stacking Cranes (ASCs). In this field, Europe Combined Terminals (ECT) in Rotterdam

Fig. 6.7.1 Automatic stacking crane

is well known. Since 1990, eight huge Over-Panamax Cranes and 25 ASCs, all of them built by Nelcon – Rotterdam, plus a great number of Automated Guided Vehicles (AGVs) transport more than 500 000 containers per year over the Delta/Sea-Land terminal of ECT.

Very few personnel are needed, in relation to the high throughput, resulting in increased efficiency. The DDE-terminal '2000-8' of ECT is now also equipped with even more and larger Over Panamax Cranes and ASCs. Transport over the terminal is also done by AGVs.

The unmanned and fully automated working stacking cranes receive their orders from a central point via a Main Computer System (MCS). This MCS tells the ASC to pick up a certain container and bring it exactly to a certain position. For the ASCs, the commanding MCS and positioning systems can be schematized as follows in Fig. 6.7.3.

Encoder systems

Incremental encoders can count very accurately the numbers of revolutions which rotating systems like the motors, wheels or measuring wheels of a crane or trolley make. The number of the counted revolutions indicates the distance over which the crane or trolley has travelled. However, if there is slip or 'creep', the measurement is no longer accurate.

Fig. 6.7.2 AGVs and ASCs

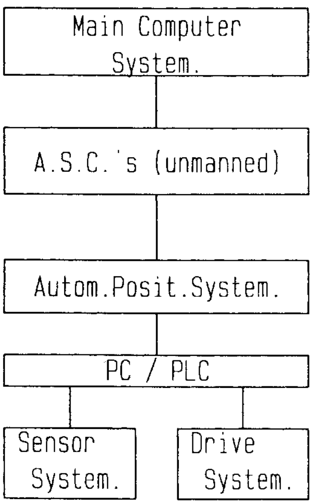

Fig. 6.7.3 The main computer system

Therefore, absolute setting points along the track are necessary. These setting points check the precise position of the crane and are used as a resetting point. These absolute setting points can for instance be detected via flags. These flags are positioned exactly along the track. An infrared sensor on the crane detects the flags.

Sensor systems

Only a few field-proven and fool-proof systems are discussed.

Hall magnets with electronic measuring rulers

The stacking area of an ASC (Automated Stacking Crane) is divided into blocks of 3,25 metres. A container of 20 feet (length 6,05 m) needs two blocks; a container of 40 feet (length 12,1 m) needs 4 blocks. Also 45 feet containers can be stacked.

As each second Hall magnet is laying on a different distance from the end, each block has its unique distance ($L_2 - L_1$), through which the Hall sensors can identify the exact position via the PLC in the crane.

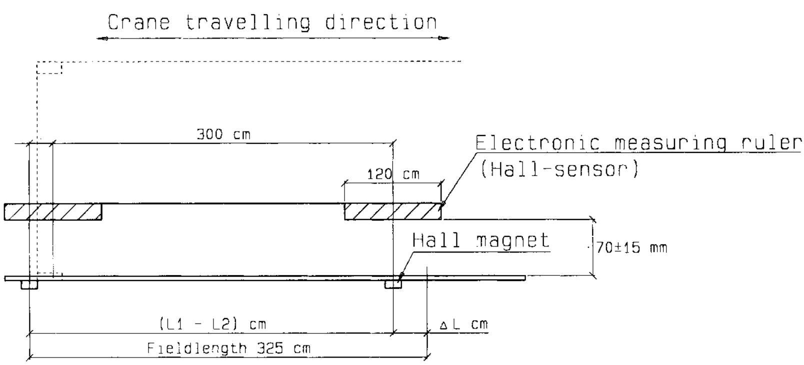

Fig. 6.7.4 Container positioning with Hall sensors

Detectors with linear absolute encoders

In this case, the build-up of the stack is somewhat more flexible. The route of the ASC is not divided into blocks. So-called Omega profiles, each with a length of 2,33 metres, are filled with small magnets which give a unique response to the signals of the measuring positioning detector, which is fastened to the crane that is running over the rail-track. This detector is connected to the PLC in the crane, and indicates the

Fig. 6.7.5 Detection with Stegmann Omega profiles

position of the crane accurately, due to the Omega profiles which lay over the full length of the crane-track. (Patented by Stegmann.)

Antenna–transponder systems

In such a system electromagnetic radiosignals are used as well as a tachometer system; the sender/receiver is mounted in an antenna on the crane. The sender transmits a signal down to a precise line on the railtrack. A number of transponders are installed along the full length of the railtrack. When the crane runs over a transponder, the transponder receives the signal from the sender, and reflects a unique signal back towards the antenna. This unique signal indicates the exact position of the crane.

The antenna sends the unique signal to the extra PLC, which decodes the signal to the position of the crane and can send information on to the main computer. The antenna also measures the relative distance Δy between the centre of the antenna and the transponder. The exact position of the crane is then $y_1 = y + \Delta y$.

The small transponders bedded-in along the whole crane- or trolley-track, are the fixed points over which the crane with the antenna can fix its position, within a very small tolerance. This system should be immune to radio-disturbances; however, a nearby high-tension or

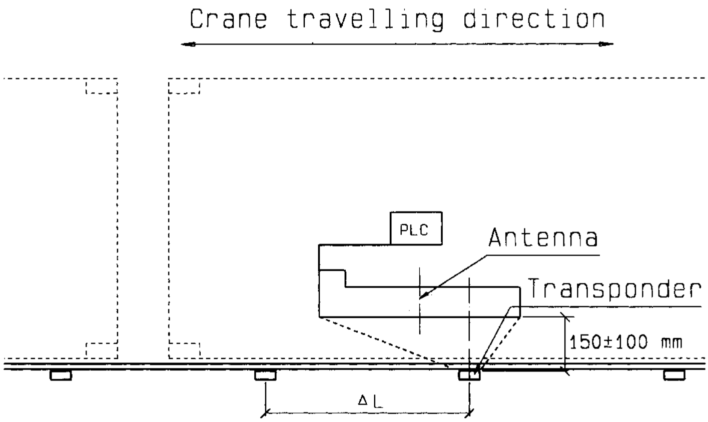

Fig. 6.7.6 Antenna–transponder system

medium-tension feeding cable of a crane can influence the working of the system.

Radar systems

A radar system on a terminal can send out radar waves to, for example, an Automated Guide Vehicle in order to guide and position this AGV.

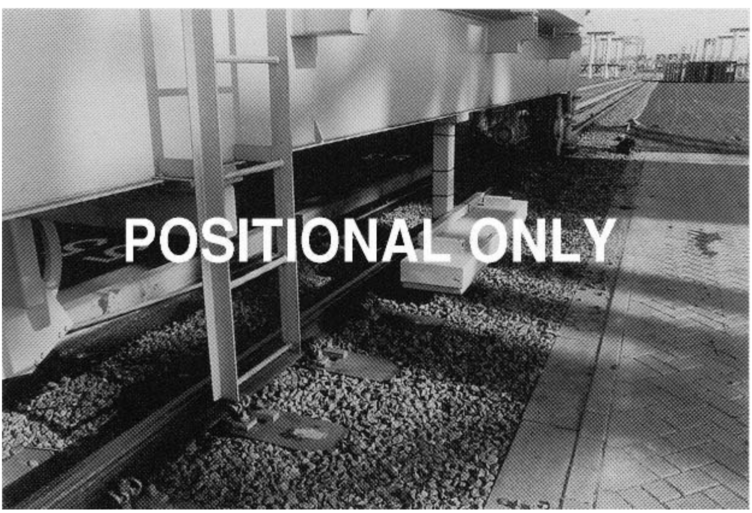

Fig. 6.7.7 Antenna block for transponder system

Laser systems

Laser beams can also be used for exact positioning. Fog, dirt and the travelling distance can influence the accuracy of the positioning. On a stacking crane, a laser beam can be positioned above each sillbeam, giving a horizontal laser-beam parallel to the crane track. The two laser beams can also then control the exact length that each crane leg system has travelled; thus checking the skew of the crane. In case of skewing too much, the laser system blocks the crane travelling mechanism, after this resetting has to be done.

If a laser camera is mounted on a trolley, with its laser beam directed vertically downwards, this system can be used to detect the distance of the trolley from stacked containers or to sillbeams etc. After a practice run this system can then be used as a way of measuring and detecting the protected areas underneath a crane.

The influence of wind and eccentric loading of the container

The measures, mentioned above indicate some of the methods by which the exact position of the crane and trolley can be found. However, this does not necessarily mean that the container is placed in exactly the right position. If there is a strong wind pushing the containers aside, or

Fig. 6.7.8 Nelcon ASCs with anti-sway system

a large eccentric load in the container, the crane will stop at the accurate position, but the container will still be incorrectly positioned. To be able to cope with the wind and the eccentric loading a strong Anti-Sway system is necessary.

The prevention of skew

The crane travelling mechanism has built-in pulse-counters which sense each rail. These pulse-counters check the distance which the crane legs have travelled over each rail, using the number of revolutions that each pulse counter has made. The PLC on board the crane compares the number of revolutions of each pulse counter. When the difference becomes too large, the crane will stop. Placed at 40 to 50 m intervals there is a position bar alongside each rail which acts as a checkpoint. An effector on the crane above each rail then checks the actual skew of the crane. The crane-driver can reset the crane and diminish or delete the skew of the crane at any checkpoint. Slip or creep of the wheel can make resetting necessary.

6.8 The automatic identification of containers

These systems belong to the Automatic Equipment Identification (AEI). The AEI systems are based on a Radio Frequency (RF) technology and consist of programmable tags or transponders, which are installed on (moving) equipment as containers, AGVs, trailers, etc. as well as Electronic Sensing Equipment or a Reader System which are placed in strategic locations like terminal entrances/exits etc.

Tags

The tag, of normal dimensions approximately $250 \times 50 \times 15$ mm, contains an antenna, a programmable microchip, a battery with a lifetime of some ten years, and a specialized switch. The reader system can read the codes of the tag up to a distance of approximately 13 m, even when the equipment on which the tag has been fastened is moving along with a speed of approximately 50 km/hour. The tag must pass through the beam from the reader system. The antenna of the tag picks this beam up, and then activates the sender switch, which sends its unique codes back to the reader. The reader decodes this signal and sends them on to the computer system.

ISO 10374 (First edition 1991 – 10 – 01) specifies a system for the automatic identification of containers. This is useful reading for those wishing to find out more about the technical specification for this system.

INTERNATIONAL STANDARD

ISO 10374

First edition
1991 – 10 – 01

Freight containers – Automatic identification

Conteneurs pour le transport de marchandises – Identification automatique

Introduction

This International Standard specifies a system for the automatic identification of freight containers and the electronic transfer of the identity of the container and permanent related information to third parties in a standard format. It is intended that the Automatic Equipment Identification (AEI) system will facilitate documentation, resource control, and communications (including electronic data processing systems). The visual container identification markings specified by ISO 6346 are not affected. Future additions to this International Standard will specify modulation, encoding and an open protocol.

Annex B, which is an informative annex only, describes the technical specification of a system that complies with the requirements of this International Standard. Parts of annex B are covered by patents held by:

Amtech Corporation
17304 Preston Road E 100
Dallas, Texas 75252
USA

The patent-holder has stated that licences will be granted under reasonable terms and conditions.

4 Operational requirements

4.1 Basic components of the AEI system

The AEI system shall consist of two basic components, i.e.

(a) an electronic device (tag) installed on the freight container, and
(b) electronic sensing equipment located apart from the freight container.

4.1.1 The tag shall be capable of

(a) maintaining the integrity of the freight container identification and permanent related information,
(b) encoding its information into a form suitable for conveyance to sensing equipment,

(c) being programmed in the field; however, the permanent data shall not be reprogrammable while the tag is fixed to the container,
(d) being physically and electronically secure and tamper-proof,
(e) being fixed to a container in accordance with the provisions of the TIR convention,
(f) being mounted between the typical vertical ribbing of the freight container side – it shall have dimensions as small as possible but not exceeding 30 cm × 6 cm × 2 cm,
(g) a minimum life of 10 years normal operational use and shall not require periodic maintenance,
(h) providing, through the sensing equipment, an indication of impending battery failure if it contains a battery,
(i) being read when it is:

   (1) operated in the environmental conditions specified in 4.6.1,
   (2) within range (see Table 1),
   (3) moving in relation to the sensing equipment at an acceptable speed (see Table 1),
   (4) sufficiently separated from adjacent similar tags to allow discrimination (see Table 1), and
   (5) suitably oriented (see 4.6.2), and

(j) international operation, without the necessity of licensing tags individually.

4.1.2 The sensing equipment shall be capable of

(a) reading information contained in the tag when it is properly presented, and
(b) decoding the information contained in the tag into a form suitable for transmission to automatic data processing systems.

4.2 Information content of the tag

The information contained in the tag is in one or more of the following categories:

- mandatory, permanent (not-changeable) information;
- optional, permanent (non-changeable) information;
- optional, non-permanent (changeable) information.

Optional information contained in a tag shall not adversely affect the operation of systems requiring only the mandatory information contained in the tag.

4.6.4 AEI system reliability and accuracy

Tags which are positioned, programmed and presented to the sensing equipment in accordance with the provisions of this International Standard shall have a minimum AEI system reliability of 99,99 percent, i.e. no more than one no-read event in 10 000 readings, and an AEI system accuracy of 99,999 9 percent, i.e. one undetected incorrect reading in 1 000 000 readings.

4.6.5 Tag positioning

The tag shall not protrude beyond the envelope of the freight container and shall have provisions for permanent mounting that will not render the structural or environmental integrity of the freight container below the service requirements.

Only a single tag shall be required to identify each freight container. The tag shall be located on the exterior surface of the right sidewall as seen from the door end of the container, approximately 0,3 m from the edge with the blind end, in the case of containers of length 12,2 m (40 ft) or less (recessed between the first and second sidewall corrugations, if applicable) or approximately 0,3 m to the rear of the lifting position, in the case of containers of length greater than 12,2 m (40 ft), but, in any case, not more than 5,94 m from the transverse centre-line of the container (see Figs 3 and 4). The letter *h* indicates the overall container height.

In the case of non-box, smooth-skin and thermal containers, the tag may be located in proximity to the lower corner fittings as an alternative to the preferred location described above.

6.9 Electronic Data Interchange (EDI)

General

Many systems are now available for:

- Ship planning
- Berth planning
- Yard planning
- Positioning of Automated Guided Vehicles (AGVs)
- Positioning of Straddle Carriers
- Positioning of containers in a stack, etc.

These systems can be regarded as belonging to the Electronic Data Interchange sector (EDI). Radio Data Equipment is frequently used, as well as a Global Positioning System (GPS), which is a satellite-based location system.

GPS and DGPS

These Global Positioning Systems and Differential Global Positioning Systems are based on the use of satellites and are among others used for locating and guiding straddle carriers on a terminal. The most widely used satellites provide signals with which, for instance, a straddle carrier position can be given with an absolute accuracy of some 50 metres using GPS which is now declassified from military only uses. By correcting the data from additional navigation equipment, situated in a reference station, the accuracy of this so-called DGPS-system (Differential Global Positioning System) can pinpoint location to approximately 1 m in 94 percent of the cases. This positioning information can be transferred to the central container management system. DGPS can also be used to prevent collision between carriers, etc. If two carriers are moving forward to each other, the DGPS detects this and gives a warning. The same system could also be used for AGVs.

Fig. 6.9.1 DGPS antenna on a straddle carrier

Automatic vehicle and container location

If a truck or container is equipped with a receiver/communication data terminal (dimensions approximately $25 \times 18 \times 4$ cm), the position of the truck or container can be traced via GPS or DGPS and contact can be kept through this with the mother company.

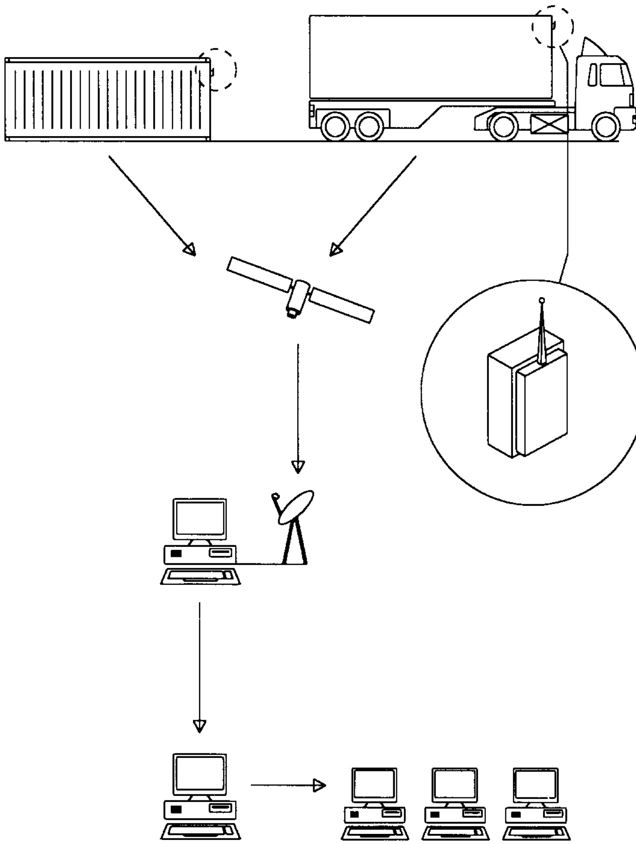

Fig. 6.9.2 Position of a container scanned by DGPS

Scanning

Scanning is popular to identify container and mass-products. It is normally based on RDF (Radio Data Frequency). For example, on an object a high quality bar code or tag is mounted. This bar code or tag can be read with the scanner, which can show the read-out code on a screen. The operator can use his keyboard to type in all the necessary

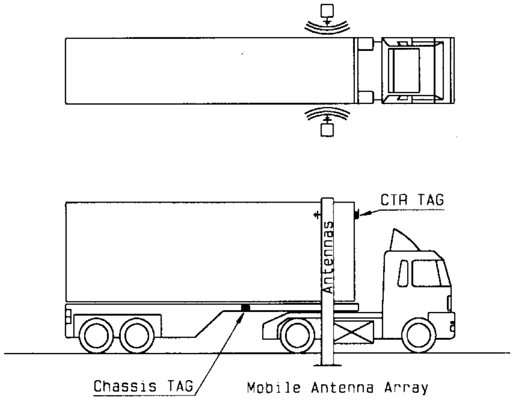

Fig. 6.9.3 Tagging and scanning

data; these data can then be transferred to the network controller, which can store the details in its memory and also can activate other RDF terminals to take action.

The scanners can be hand-held, vehicle mounted or mounted in a stationary position. However, in order to have an accurate reading the scanner must not be further than about 6 metres from the tag. The reading width of the scanner can be from maximum 1 to 2 m, depending on the distance from the code. For this type of optical identification, readers can be used which can read:

– bar codes;
– dot codes;
– optical colour codes.

Radio Wave Identification systems and Inductive Identification systems have a Read and Write Unit which can read tags over a distance of approximately 0,7 m to 55 m. Tags which have been mounted on a truck or container in a standardized position can be read by a reader, situated at the gate of a terminal. The reader can pass the information directly to the desk of the gate clerks, thus saving them time, and improving security.

Magnetic cards encoded with for instance, information about the driver, his licence number, or information about the container can also be used. Magnetic card readers are then used to read the available information on these magnetic cards.

Container inspection by X-ray

An X-ray scanning system can be mounted in a terminal or harbour entrance. Customs can use X-ray scanning systems to control what is being transported in a container. Some 50 trucks per hour can be controlled by such an X-ray system.

Seal recognition

The seal of a container should be checked. This must be carried out manually to see whether the seal is intact. A video recorder can also be used, although this is less efficient.

CSC plate control

On a container a so-called CSC plate is fastened. This plate is the so-called Container Safety Certificate and gives information about the maximum allowed weight and the date up to which the container is under the cover of insurance.

If it is necessary to control this plate, it is normally done by an official who has to look carefully at the often dirty and corroded CSC plate. This requires special attention and takes time. Trials are being carried out with special cameras to try to automate the checking of CSC plates.

Fig. 6.9.4 Checking with a hand-held terminal

Checking the damage to containers

Checking the actual damage to a container can be done by an official; but also some cameras can be mounted, which film the container from various angles in order to fix the 'outlook' of the container and to provide evidence of any damage.

6.10 GE Toshiba Automation Systems: crane automation

GE Toshiba Automation Systems of Salem, Virginia, USA has, like other electrical system manufacturers, developed complete crane automation systems. GE Toshiba uses laser and infrared sensors combined with pulse counters, etc. in their system.

The following operational features are used:

- Sway Position Feedback System (SPFS)
- Automatic Position Indication System (APIS)
- Profile Scanning System (PSS)
- Automatic Landing System (ALS)

Sway Position Feedback System (SPFS)

The sway angle displacement and the skew angle of the spreader are measured relative to the trolley. An infrared sensor is located under the trolley and monitors the positions of three infrared light sources on the headblock/spreader. The sway angle and skew angle measurements are used in the image processing algorithms, as well as for the PSS and ALS.

Automatic Position Indication System (APIS)

To check the *trolley position*, the position instrument calculates the trolley position with a digital tachometer/encoder which is connected to a non-driven trolleywheel. The trolley position is continuously recalibrated to a certain position of the trolley travel route by multiple trolley recal-flags and a 'trolley near-home' switch.

An absolute position check is also given by a laser rangefinder, which is located in the backreach of the trolley in a fixed position. The rangefinder laser beam is aimed at a large reflector on the trolley. This rangefinder can provide a distance measurement accuracy of plus and minus 20 mm at 120 m distance. In a similar fashion, the infrared sensor, located under the trolley provides an absolute position check on the main-hoist encoder position, tracking to an accuracy of plus and minus 20 mm.

For the *crane position* the same system can be used as for the trolley position. However, a transponder system can also be used for exact crane position detection. These transponders are bedded-in along the cranetrack(s). Each transponder identifies a unique crane position, which can be read by any of the cranes which pass over the transponder with its transponder-antenna. The transponder- and interpreter unit also generates a midpoint pulse that is symmetrical about the exact centre of each transponder. These pulses are used as recalibration flags such that the position calculated and stored in the drive is updated at each transponder crossing.

The main hoist position is established and monitored in a similar fashion to the trolley motion using feedback from encoders coupled to the motor shaft. However, the home or recalibration flag is based on a block-operated limit switch located within the normal upper slowdown zone. A laser range finder can also be used as a primary or redundant back-up position feedback device. Based on the main hoist position instrument in the drive, automatic positioning can be realized through the LAD (Look Ahead Distance) software algorithm.

The LAD system also incorporates the end-limit switch system, sillbeam protection, etc. as well as automatic positioning to a point. LAD applies to all crane motions!

Profile Scanning System (PSS)

In the PSS a special long range, high accuracy, non-reflective laser sensor is used. It monitors the height of each row in the bay of a vessel or each slot in the stacking area if it is applied to a stacking crane.

The stack profile is monitored continuously and dynamically as the trolley or crane moves over the stacks during the intended container moves. In this way the clear height for the most efficient movement over the container stacks can be determined and the continuous movement of ships, the tidal flows, the ship's ballast adjustment, etc. can be followed.

By means of the PSS the clear heights can continuously be updated to achieve minimum cycle times and to avoid collisions. Further anti-collision systems are also employed for safety reasons.

Automatic Landing Systems (ALS)

One of the keys to crane productivity lies in the final landing process. Laser technology is used for the edge detection of the spreader, the container and the chassis or AGV. The ALS can include a docking status error check. It can maintain near-perfect container stacks in the stacking area and it could be used to assist the operator in landing the

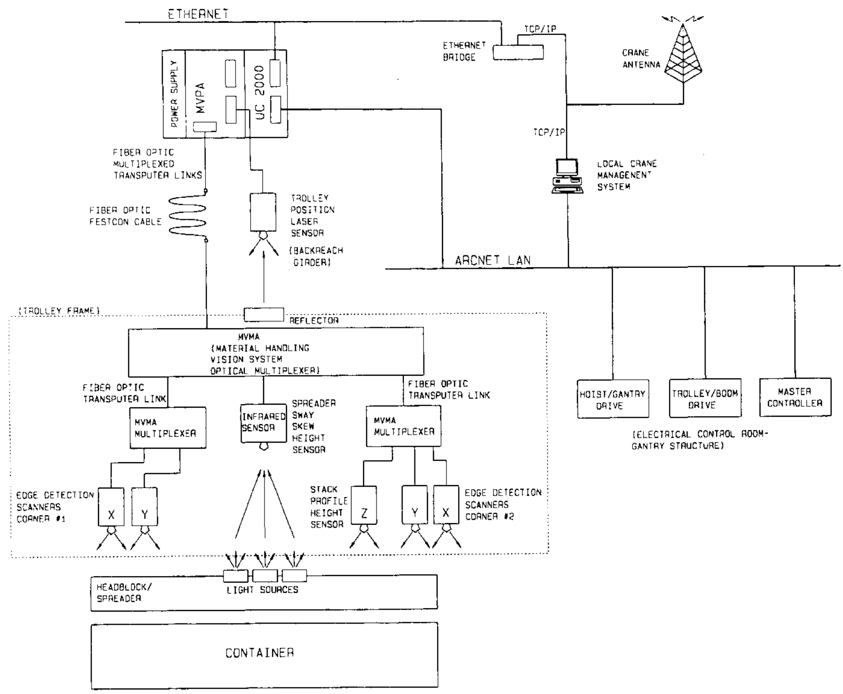

Fig. 6.10.1 Hardware topology

spreader onto the twistlock castings which are used to lift the hatch-covers of a ship.

Sensors

The following sensors are used on a container quay crane:

- An infrared sensor system underneath the trolley for measuring the sway angle in the direction of trolley travel and the spreader skew angle (SPFS). This sensor may also be used to periodically verify the main hoist position as a redundant check on the primary position instrument in the drive.
- Four two-dimensional (X and Y) laser scanners on the trolley are used to locate the corners of the spreader, container, chassis or AGV with an accuracy greater than plus or minus 20 mm on 35 m distance.
- Each of the two corner units contains two high speed laser scanners which measure the exact location of the corners of the spreader,

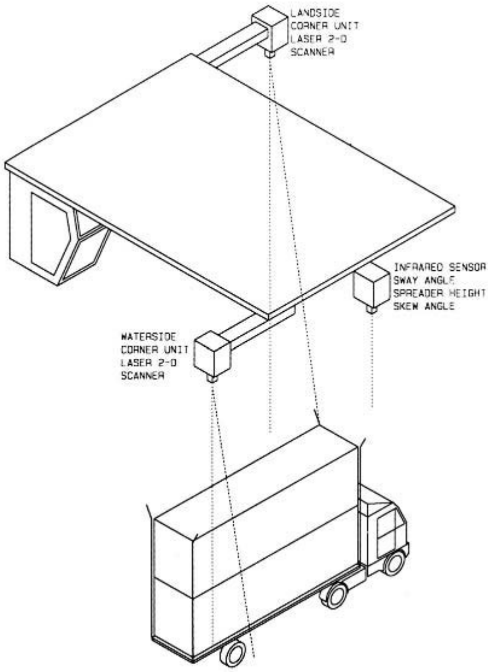

Fig. 6.10.2 Sensor arrangement on a trolley of a quay crane

container, chassis or AGV. One corner unit monitors the waterside corner and the other the landside corner. The waterside corner unit also includes the laser range finder which monitors the stack profile and it also can peer into the hold of the ship.

A laser range finder is mounted underneath the trolley and is used for stack profiling. It measures the distance without the use of dedicated reflectors, and can achieve an accuracy of plus and minus 10 mm at distances of up to 40 m (PSS).

– A laser range finder is mounted on the backreach of the crane for an absolute position check of the trolley (APIS).

Learn cycles

First, the crane driver has to handle a container in a normal manner manually, before the automatic mode is used. (Where anti-collision devices are not used, Overhead Bridge Cranes (OHBCs) will not require a learn cycle.)

Chassis Alignment Systems

Special camera-based vision systems may be employed to enhance the procedure of aligning the transportation equipment (truck chassis, bump-car, road chassis, etc.) under the crane. The vision-type imaging system may use natural or artificial light sources or structured light sources. Based on the image processing, the system indicates to the truck chassis driver when the chassis or container is aligned with the centreline of the crane. In addition, the same image processing is utilized to automatically position the trolley over the chassis/container and automatically adjust the skew angle of the spreader to match that of the chassis/container. The system indicates to the crane operator when the chassis position is outside the allowable skew angle which would allow landing of the spreader. The crane is equipped with a visual signalling system that indicates to the truck driver when he is approaching the correct position, when to stop, and when he has overshot the correct alignment position.

Container Recognition Systems

Camera-based imaging systems are also employed to automatically recognize the identification numbers printed on the sides and/or ends of the containers moved in the terminal. As the operator moves the container to or from a vessel, cameras located at multiple locations on the crane capture the ID number. The ID number is then passed over a data network to a Yard Management System for processing. The Yard Management System then issues orders or instructions to the yard transportation equipment for proper dispatch of the container.

Yard Management Systems

The cranes may be supplied with wireless RF, optical fibre, or waveguide communication technology to interface with a Yard Management System. The Yard Management System directs the movements of the ground traffic, yard stacking crane, and the ship to shore cranes. The sensors used on board the cranes for the various automation functions discussed previously, are utilized to establish reports to the Yard Management System that include container size, container weight, container pick-up coordinates, container drop-off coordinates, twistlock positions, etc.

Acknowledgement

Source of information for Section 6.10, Mr John T. Sholes, GE Toshiba Automation Systems, Salem, Virginia, USA.

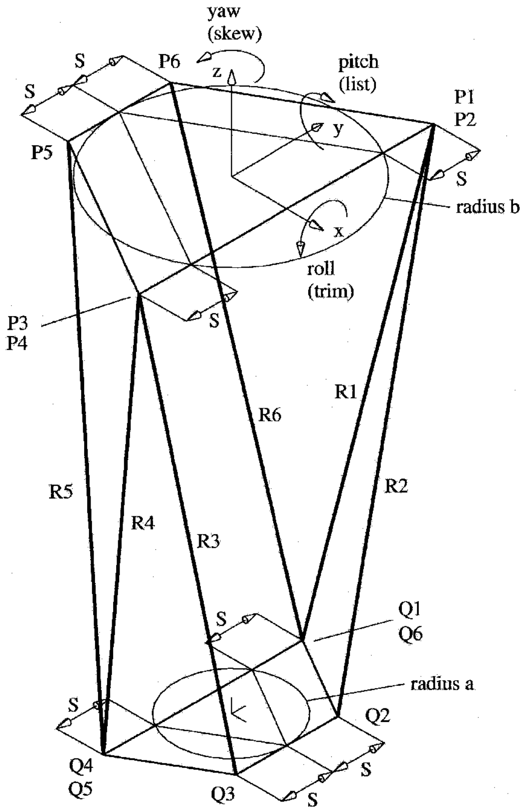

Fig. 6.11.1 Definition of the geometry

6.11 The Stewart Platform Reeving

Patrick Stevedores Pty Inc. and the Australian Centre for Field Robotics (ACFR) at the University of Sydney have recently developed a fully patented reeving system for the hoist mechanisms of container cranes: the Stewart Platform Reeving. Figure 6.11.1 shows the schematic layout of the reeving, while Fig. 6.11.2 shows the 1:15 scale working model of the installation.

The figures show the six hoisting wire ropes of the system. When the six wire ropes are independently controlled, the six spatial degrees of freedom can be used for complete control of the load by 'micropositioning'. The reeving system then gives an excellent stiffness; trim, list, and skew can then also be implemented in the system.

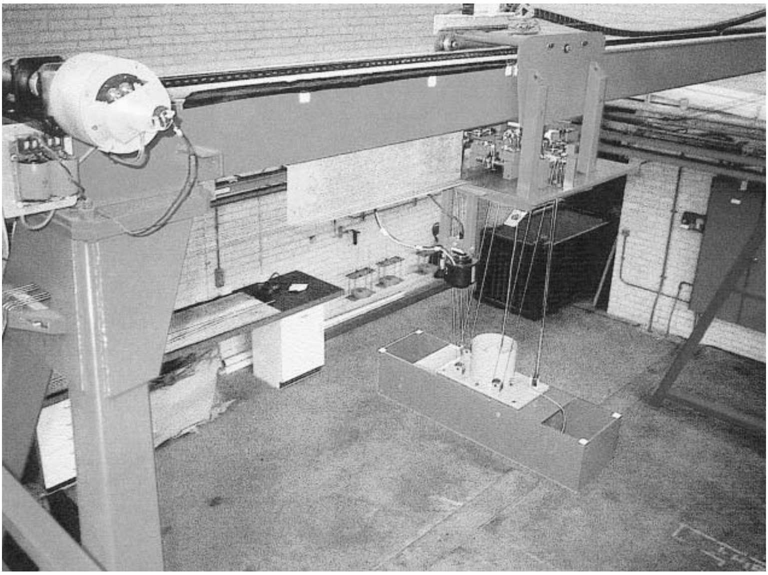

Fig. 6.11.2 The 1:15 scale working model

Stewart Platform reference:
D. Stewart A platform with six degrees of freedom. *Proc. Instn Mech. Engrs* (*London*), *Part I*, 1965, **180**(15), 371–386.

6.12 Checking the alignment of containers etc. with Laser Scanners

Lase GmbH Industrielle Lasertechnik of Bremen, Germany developed a fully patented Laser Scanning System with which the distances between the spreader and a container, an AGV or trailer, etc., as well as the relative position of these parts to each other can be measured.

When setting down a container on to – or taking a container from – an AGV, the misalignment between the container and the AGV, as well as the relative distances between spreader and container or AGV can be controlled by using the Rotating Laser Scanners, in combination with an Evaluation Unit, being a supervisory PC. If misalignments are indicated, a crane driver or checker can correct the positioning.

Figure 6.12.1 gives an overview of the system; Fig. 6.12.2 shows the measuring and positioning of the scanners at some 22 m above the quay level; and Fig. 6.12.3 shows one of the ZPMC cranes, equipped with this system.

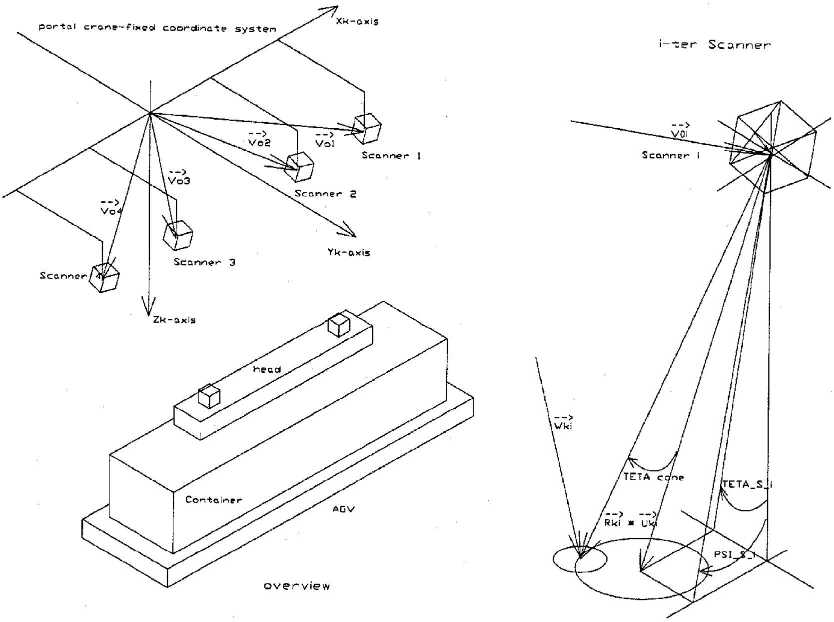

Fig. 6.12.1 Overview

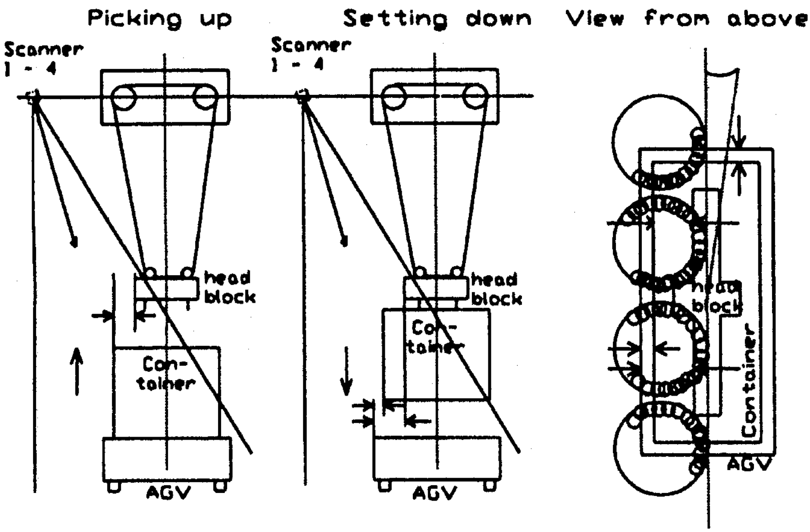

Fig. 6.12.2 Positioning of the scanners

Fig. 6.12.3 Crane with laser scanners

The scanners have a cone-shaped scan which gives a cone diameter of approximately 2,8 m over 20 m height. The cone-circle is scanned in steps of 0,25 degrees.

6.13 Spreader Positioning System

Nelcon's '1 over 4' or 5-high Automated Stacking Cranes (ASCs) are provided with a special 'rope tower' wire rope device for the hoisting mechanism. Due to the way of reeving and the lay-out, this rope tower is very stiff and permits the (fully automated) ASC to stack the containers accurately on top of each other.

However, under extreme conditions, e.g. a heavy storm, it is possible that the required stacking accuracy cannot be met. For this purpose BTG Engineering BV in Maasdam developed a fully patented Spreader Positioning System, which controls the eventual sway or swing of the spreader.

The spreader itself is therefore provided with hydraulic cylinders, which are controlled by the PLC in the crane. On the spreader a box is

Fig. 6.13.1 ASCs with Spreader Positioning Systems

Fig. 6.13.2 Box with PSD chip

mounted in which a LED system throws a beam of light towards a box underneath the trolley. In this box a PSD chip (Position Sensitive Device Chip) and a special lens are mounted. The beam of the LED system is guided through the lens and hits the PSD chip. Through this chip the PLC gets the various commands to activate the hydraulic cylinders on the spreader, thus forcing the spreader with the underhanging container to change its position.

6.14 Camera-Monitor Systems

Orlaco Products BV in Barneveld, The Netherlands, manufactures camera systems which help crane drivers, truck drivers, etc. to overview

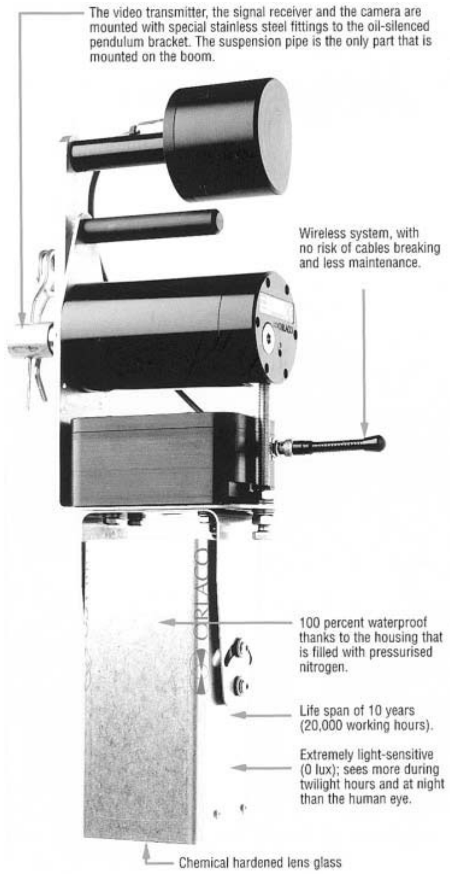

Fig. 6.14.1 The camera system

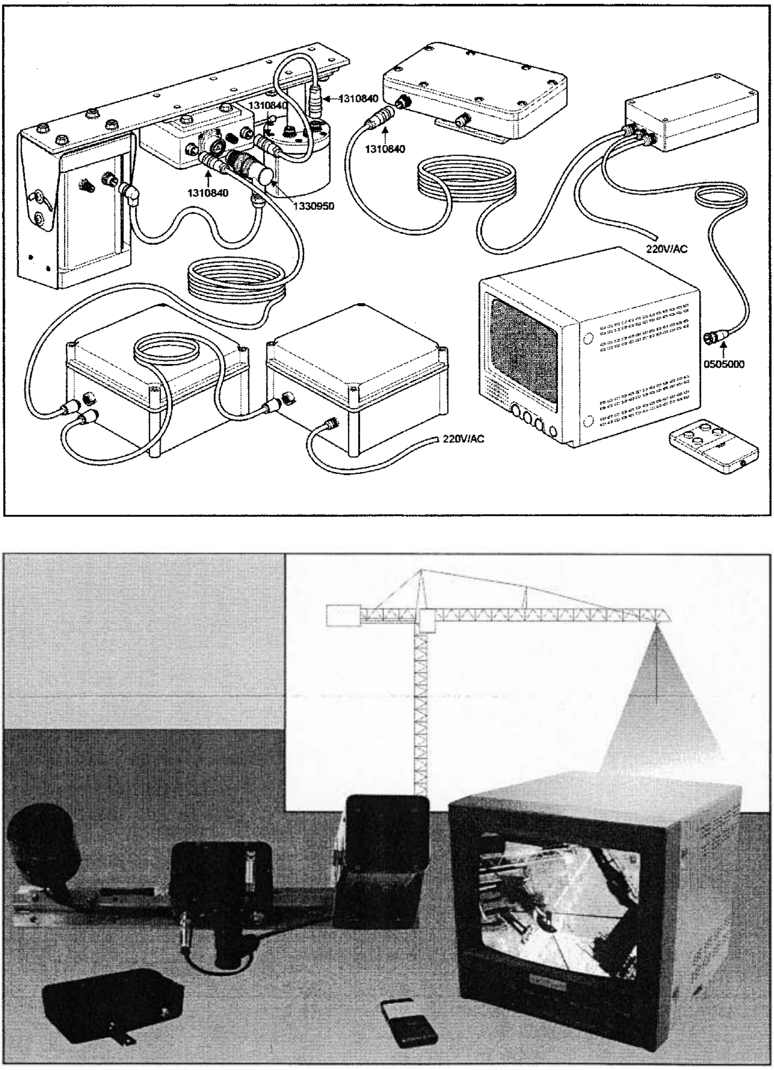

Fig. 6.14.2 Overview of the system

the work area, the winches, etc. The camera of such a system is shock and vibration proof and is extremely light sensitive. The system gives the crane driver a much better feel for his work. As the lenses of the cameras are heated, condensation and frost have no influence on the camera.

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Chapter 7

Construction and Calculation Methods on Strength and Fatigue

7.1 Materials

(A) For steel constructions

Table 7.1.1 Table of corresponding former designations

Designation		*Equivalent former designation in*				
Acc. to EN 10027-1 and ECISS IC 10	Acc. to EN10027-2	Acc. to EN 10025: 1990	Germany	France	United Kingdom	Older standards
S185	1.0035	Fe 310–0	St 33	A33		Fe 320
S235JR	1.0037	Fe 360 B	St 37–2	E 24–2		Fe 360 B
S235JRG1	1.0036	Fe 360 BFU	USt 37–2			
S235JRG2	1.0038	Fe 360 BFN	RSt 37–2		40 B	
S235JO	1.0114	Fe 360 C	St 37–3 U	E 24–3	40 C	Fe 360 C
S235J2G3	1.0116	Fe 360 D1	St 37–3 N	E 24–4	40 D	Fe 360 D
S235J2G4	1.0117	Fe 360 D2				
S275JR	1.0044	Fe 430 B	St 44–2	E 28–2	43 B	Fe 430 B
S275J0	1.0143	Fe 430 C	St 44–3 U	E 28–3	43 C	Fe 430 C
S275J2G3	1.0144	Fe 430 D1	St 44–3 N	E 28–4	43 D	Fe 430 D
S275JG4	1.0145	Fe 430 D2				
S355JR	1.0045	Fe 510 B		E 36–2	50 B	Fe 510 B
S355J0	1.0553	Fe 510 C	St 52–3 U	E 36–3	50 C	Fe 510 C
S355J2G3	1.0570	Fe 510 D1	St 52–3 N		50 D	Fe 510 D
S355J2G4	1.0577	Fe 510 D2				
S355K2G3	1.0595	Fe 510 DD1		E 36–4	50 DD	
S355K2G4	1.0596	Fe 510 DD2			50 DD	
E295	1.0050	Fe 490–2	St 50–2	A 50–2		Fe 480
E335	1.0060	Fe 590–2	St 60–2	A 60–2		Fe 580
E360	1.0070	Fe 690–2	St 70–2	A 70–2	A 690	Fe 690 B

Note: 1. EN = European norm (Euro Norm).

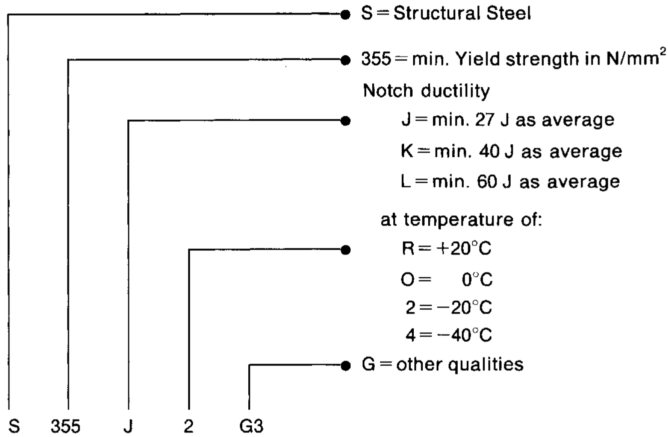


Table 7.1.2

Designation Acc. to EN 10027–1 and EC/SS 1C 10	Acc. to EN 10027–2	*Desoxi-dation method*	*Sub-group*	*C in % for nominal plate thickness in mm* < 16	> 16	> 40	M_n *% max.*	*Si % max.*	*P % max.*	*S % max.*	*N % max.*
S 185	1.0035	Choice	BS	—	—	—	—	—	—	—	—
S235JR	1.0037	Choice	BS	0,17	0,20		1,40	—	0,045	0,045	0,009
S235JRG1	1.0036	FU	BS	0,17	0,20	—	1,40	—	0,045	0,045	0,007
S235JRG2	1.0038	FN	BS	0,17	0,17	0,20	1,40	—	0,045	0,045	0,009
S235JO	1.0114	FN	QS	0,17	0,17	0,17	1,40	—	0,040	0,040	0,009
S235J2G3	1.0116	FF	QS	0,17	0,17	0,17	1,40	—	0,035	0,035	—
S235J2G4	1.0117	FF	QS	0,17	0,17	0,17	1,40	—	0,035	0,035	—
S275JR	1.0044	FN	BS	0,21	0,21	0,22	1,50	—	0,045	0,045	0,009
S275JO	1.0143	FN	QS	0,18	0,18	0,18	1,50	—	0,040	0,040	0,009
S275J2G3	1.0144	FF	QS	0,18	0,18	0,18	1,50	—	0,035	0,035	—
S275J2G4	1.0145	FF	QS	0,18	0,18	0,18	1,50	—	0,035	0,035	—
S355JR	1.0045	FN	BS	0,24	0,24	0,24	1,60	0,55	0,045	0,045	0,009
S355JO	1.0553	FN	QS	0,20	0,20	0,22	1,60	0,55	0,040	0,040	0,009
S355J2G3	1.0570	FF	QS	0,20	0,20	0,22	1,60	0,55	0,035	0,035	—
S355J2G4	1.0577	FF	QS	0,20	0,20	0,22	1,60	0,55	0,035	0,035	—
S355K2G3	1.0595	FF	QS	0,20	0,20	0,22	1,60	0,55	0,035	0,035	—
S355K2G4	1.0596	FF	QS	0,20	0,20	0,22	1,60	0,55	0,035	0,035	—
E295	1.0050	FN	BS	—	—	—	—	—	0,045	0,045	0,009
E335	1.0060	FN	BS	—	—	—	—	—	0,045	0,045	0,009
E360	1.0070	FN	BS	—	—	—	—	—	0,045	0,045	0,009

Notes: BS = steel for general use; QS = quality steel.

Pipes

52 N pipes are supplied in a normalized condition. The normalizing temperature is approximately 920°C. After the required temperature has been attained over the entire cross section, the pipes are cooled off in the air. The stress relieving temperature is 530–580°C.

The 0,2 yield strength is min. 340 N/mm^2; the tensile strength min. 460 N/mm^2, the elongation is 20 percent.

(B) For mechanisms

See also Section 7.6.C; Fatigue in mechanism components as shafts; Table B1 for allowable stresses.

Materials

Fe 360	
Fe 510	CK 35
	CK 45
	CK 60
	42 Cr Mo_4
	34 Cr Ni Mo_6

7.2 Welding

Welding is one of the most important activities in the manufacture of cranes and similar equipment. A great deal of this is hand-welding; a highly skilled and precise job carried out by an experienced welder. As much as possible is done with automatic welding machines. These give a higher output and a more homogeneous weld. Widely used methods include the following.

Manual Metal Arc or MMA-welding

MMA-welding or welding by hand is mostly used for welding steel plates, angles, etc. of material quality Fe 360 or Fe 510 (S355).

The welding has to be done with basic electrodes of a high output type. The skill of the welder is most important to achieve good quality welds.

Fig. 7.2.1 The welding of a big box girder

MIG-welding

This is a gas-shielded Metal Arc welding, which is carried out using a continuous wire electrode. The continuous wire electrode is fed automatically into the welding gun, where the shielding gas is added to the welding process. The weld is then shielded by a stream of Metal Inert Gas (MIG), which is a mix of CO_2 and Argon.

Fig. 7.2.2 Welded pipe constructions

MAG-welding

Following the same process as given under MIG-welding a Metal Active Gas (MAG) is where 100 percent CO_2 is used for the shielding. The MAG process gives a rather deeper penetration than the MIG process.

Submerged arc welding or SAW-welding

In submerged arc welding an arc is maintained between the work and the end of a cored wire electrode, which is continuously fed into the arc by motor driven feed rolls. The arc is invisible and operates beneath a layer of granular flux, some of which melts to provide a protective slag cover over the weld pool.

There are hand-held welding guns available, however submerged arc welding is principally done with fully automatic equipment. It is particularly suitable for long straight joints of a very high quality, which are laid in a flat position.

Flux Core Arc-welding

This type of automatic welding can, among others, be used in the welding of pipes and tubes. The welding wire which gives slag, is of the Rutyl type and includes a small percentage of Nickel, giving a fine welding result.

Dangers: the prevention of problems

Cracks, which are of course dangerous, can occur in welds, and therefore the welds have to be controlled carefully. This can be done by visual, ultrasonic or X-ray inspection and by methods in which penetrating liquids with a magnetic control system for cracks is used.

It is always most important to avoid H_2 (Hydrogen). The carbon content of the materials must also be kept as low as possible.

The shrinkage of a construction during the welding process must be continuously controlled, and extra tensions in the construction due to shrinkage must be avoided to prevent cracks or even break-downs.

The cooling off of the weld and its surroundings must be kept within allowable limits. It is, therefore, necessary to preheat thicker constructions before welding in order to prevent the weld cooling-off too quickly in relation to its surroundings.

Welders

Welders must be well trained for their job. Their qualification goes from 1G up to 6GR.

Possible detrimental phenomena resulting from welding

Among others, these may be:

- Hydrogen cracks;
- reduction of toughness;
- solidification cracking;
- lamellar tearing – reduction of the sensitivity to lamellar tearing can be achieved by connecting all layers of the plate, see Fig. 7.2.3;
- stress relief heat treatment cracks;
- differences in chemical position, grain size and stress levels between the weld and the parent material may lead to different corrosion rates. In most cases the weld and heat affected zones are attacked preferentially;
- stress corrosion cracking.

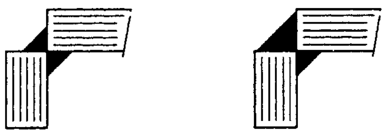


Fig. 7.2.3 Reduction of the sensitivity to lamellar tearing

Fig. 7.2.4 Submerged arc welding

7.3 Bolts

The types of bolts now widely used are 'High strength' types 8.8 and 10.9. These marks are shown on the head of the bolt and on the nut.

8.8 means: min. tensile strength: $80\,kN/cm^2$
min. yield strength: $8 \times 8 = 64\,kN/cm^2$

10.9 means: min. tensile strength: $100\,kN/cm^2$
min. yield strength: $10 \times 9 = 90\,kN/cm^2$

The plates which are bolted together must be painted with special paint on those places where the bolts are used. Sometimes injection bolts are

Table 7.3.1 Bolt characteristics

Bolt	*Tensile section*	*Tightening moment*		*Prestressing force*		*Nos. of friction surfaces*	*Transmissible shear force*		*Allowed tensile force*	
	cm^2	8.8 M kN cm	10.9 M kN cm	8.8 P kN	10.9 P kN		8.8	10.9	8.8	10.9
M 16	1,57	23	33	80	113	1	14	20	30	43
						2	28	40		
M 20	2,45	45	64	125	176	1	22	31	66	92
						2	45	63		
M 24	3,53	78	110	181	254	1	32	45	95	133
						2	64	91		
M 30	5,61	155	218	287	404	1	51	72	150	212
						2	102	144		

Fig. 7.3.1 Bolted connection

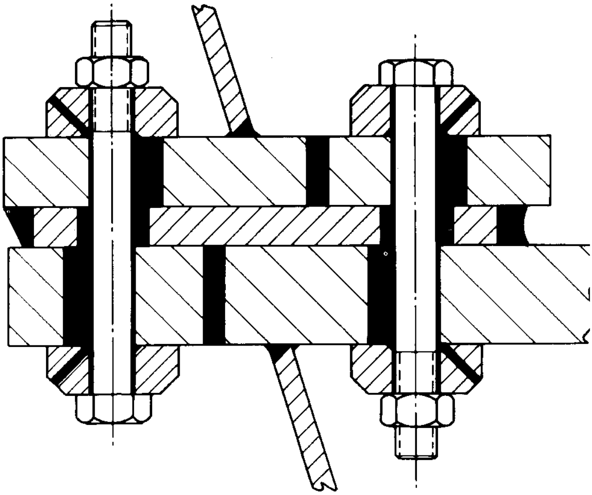


Fig. 7.3.2 Flange plates with injection bolts

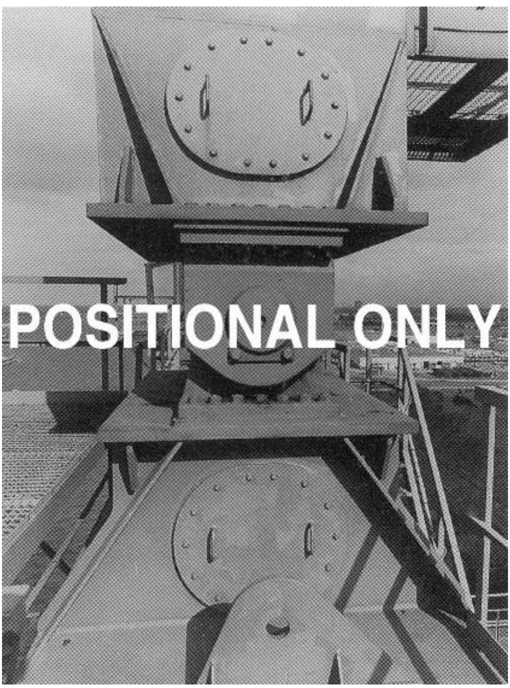


Fig. 7.3.3 A bolted A-frame construction

used in heavy construction in order to eliminate the free space which pops up when the holes in the connection plates do not mate to each other properly.

7.4 Construction of box and lattice girders, etc.

When riveting used to be the main method to connect materials, all cranes and unloaders consisted of lattice constructions. Since welding replaced riveting, box constructions have become more and more popular. Most cranes and unloaders are built up from box-type elements. However, it must be stated that welded lattice girders often can give excellent solutions for girders, booms, jibs, towercranes, etc.

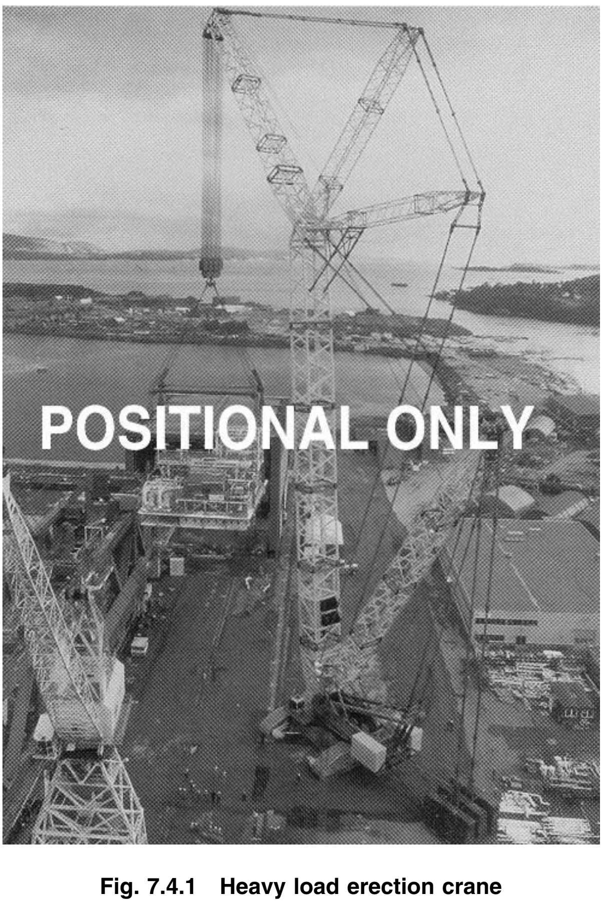


Fig. 7.4.1 Heavy load erection crane

Fig. 7.4.2 Riveted construction

Fig. 7.4.3 Bolted construction

Fig. 7.4.4 Welded construction

Fig. 7.4.5 Lattice girder construction

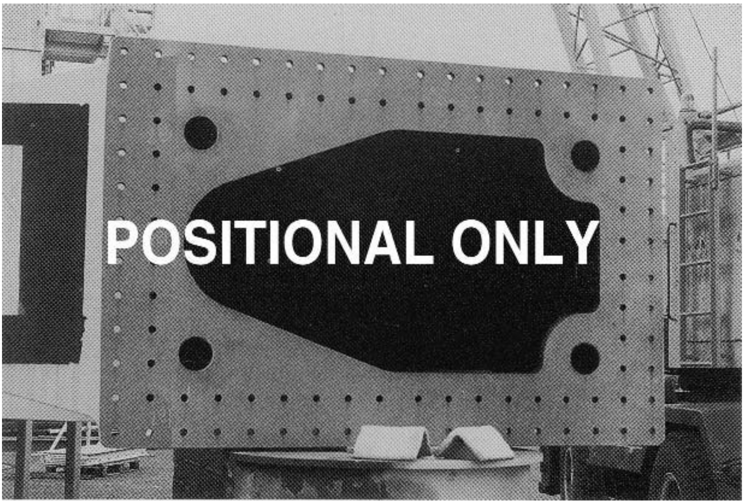


Fig. 7.4.6 Flange plate construction

Fig. 7.4.7 von Karman strips

Typical girder constructions

Figures 7.4.8 and 7.4.9 give examples of single and double box girders and lattice girder constructions.

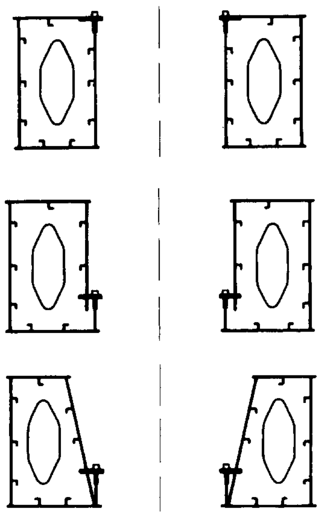


Fig. 7.4.8 Double box girder construction

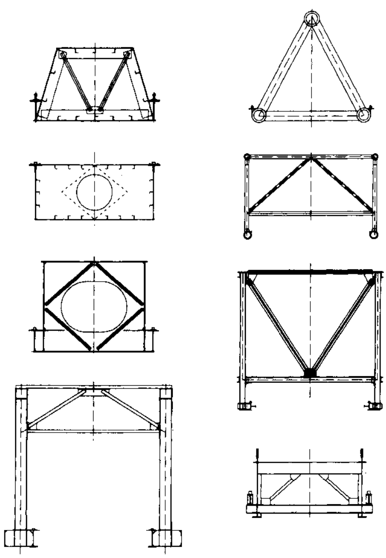


Fig. 7.4.9 Single box and lattice girder construction

Fig. 7.4.10 Lattice girder

Fig. 7.4.11 Detail of a lattice girder

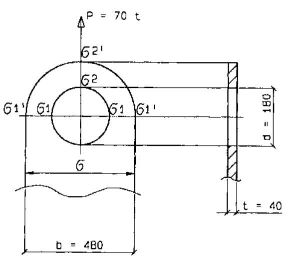
In single girder constructions the trolley hangs underneath the girder. With double girder constructions, the trolley runs above or in between the girders, thus giving a lower height crane than with the single girder construction. The legs are normally built of box girders or tubes and sill beams, mostly of box girders. These are normally welded. However for erection duties, flange plates or lap plates are used, which are bolted together. Tubular constructions must be controlled as they become subject to von Karman whirls. Wind and storm frequently cause vibrations due to these whirls in tubular constructions. Special calculation programs are available to check this phenomenon and can indicate if the vibrations are likely to become dangerous and lead to damage.

It is possible to use von Karman strips and wire ropes to dampen the shaking of the tubular constructions. These details are important. Sections 7.6 A, B, C and D on how fatigue can influence the construction; details should also be referred to. The loads placed on a construction, the quality of welds, the stress concentrations in the welds, and the stresses inherent in the construction render it more or less fatigue sensitive. The useful life of any crane construction is largely determined by these important details. Pre-heating and the control of the cooling rate of thicker plate welded constructions can also be an important feature. Care, skill and the application of skill and knowledge are vital.

7.5 Boom-ties; diagonals

Boom-ties

The stress and the stress concentrations in boom-ties must be carefully checked. Poocza offers widely used calculations for lugs. The somewhat dated calculation method of Professor Bleich gives a more conservative result, but is very convenient to use.



Example :

$$\sigma = \frac{P}{F} = \frac{70000}{48 \cdot 4} = 364 \text{ kg/cm}^2$$

$$\frac{d}{b} = \frac{18}{48} = 0{,}375 \qquad \sigma_1 = 5{,}3 \quad \text{when } \frac{d}{b} = 0{,}375$$

$$\sigma_1 = 5{,}3 \cdot 364 = 1929 \text{ kg/cm}^2$$

(Allowed is $\sigma_1 = 0{,}8\ \sigma_E$

$\sigma_E = 2600$ kg/cm² for Fe 360 ; $\sigma_E = 3600$ kg/cm² for Fe 510)

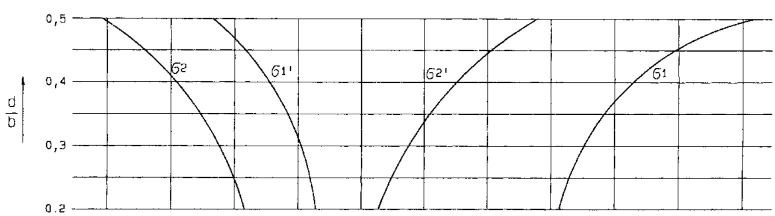


Fig. 7.5.1 The calculation of lugs according to Professor Bleich

DIN standard DIN 4133 offers the following tie-end calculation.

$$a \geq \frac{S}{2 \cdot t \cdot \sigma_{\text{allow}}} + \frac{2}{3} \cdot d$$

$$c \geq \frac{S}{2 \cdot t \cdot \sigma_{\text{allow}}} + \frac{1}{3} \cdot d$$

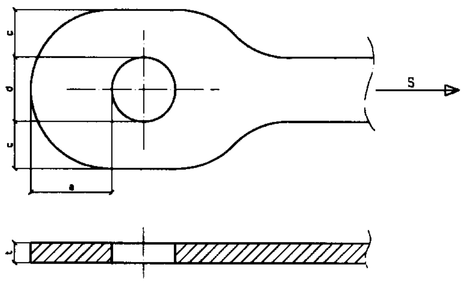


Fig. 7.5.2 Tie-end according to DIN 4133

The pressure between the pins and the bushings should be taken as:

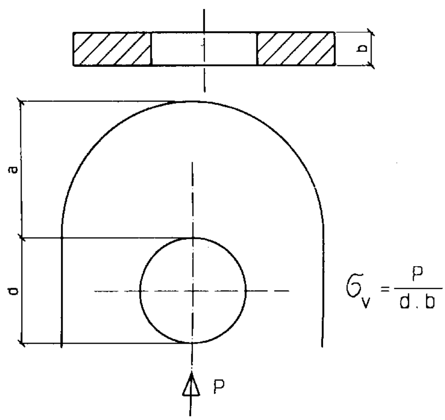


Fig. 7.5.3 Pressure between pin and bush

For bushings made of G, Ni, Al, Bz, F 68, (DIN 1714)

$\sigma_{v\,\max} = 50\,\text{N/mm}^2$ (no amplifying or dynamic coefficient to be included).

(For other bronze bushings $\sigma_{v\,\max} = 15$ to $20\,\text{N/mm}^2$.) Also glacier- or rollerbearings can be used in the hingepoints.

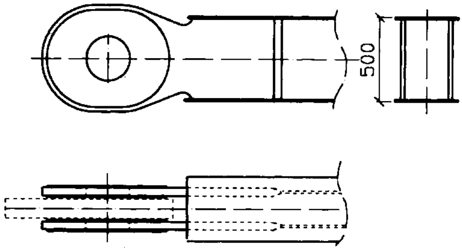


Fig. 7.5.4 Boom-tie construction 1

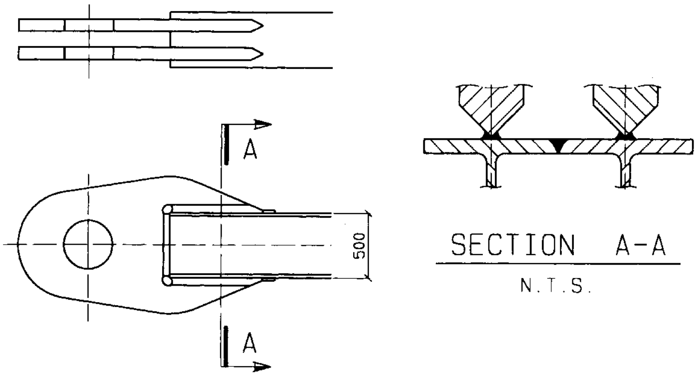


Fig. 7.5.5 Boom-tie construction 2

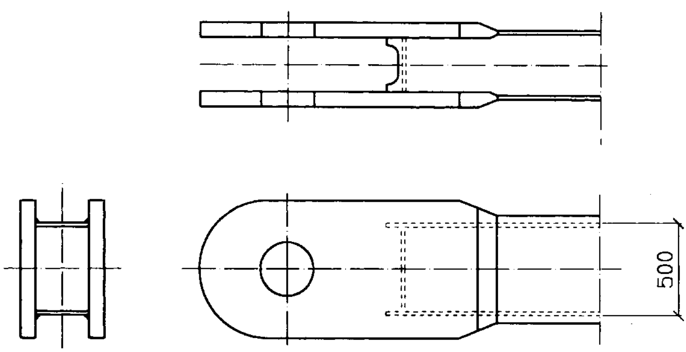


Fig. 7.5.6 Boom-tie construction 3

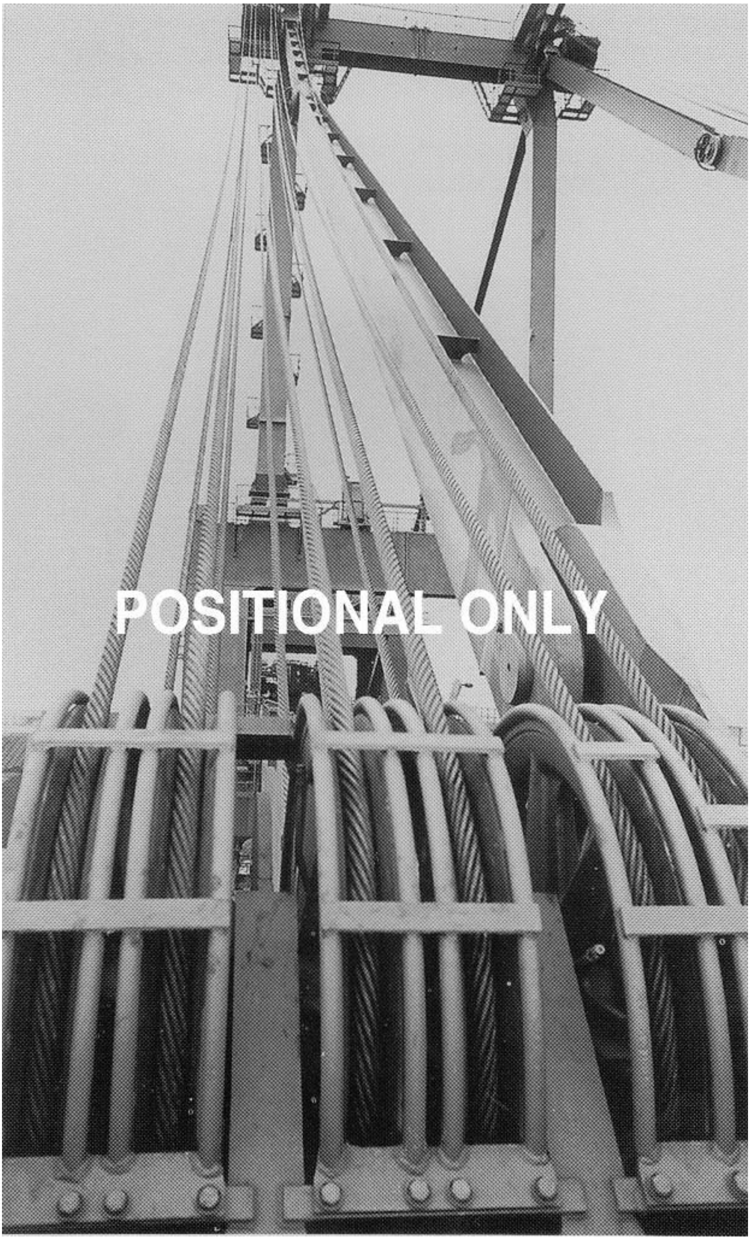


Fig. 7.5.7 Boom-tie with boom hoist tackle

The normal tensions in the boom-tie itself should be taken as approximately $0{,}6 \times \sigma_{\text{allow}}$. The Kappa factor (see the Tables in Section 7.6.B) should be calculated and the trough found as σ_{allow} should be respected.

Diagonals and back ties

Diagonals and back ties can be executed as I-profiles, box-profiles or tubes. Tubular constructions must be protected from von Karman whirls due to wind and storm. If necessary von Karman strips have to built on.

The diagonals can be connected with pins to the main constructions (see earlier – Boomties) or with welds or with bolts.

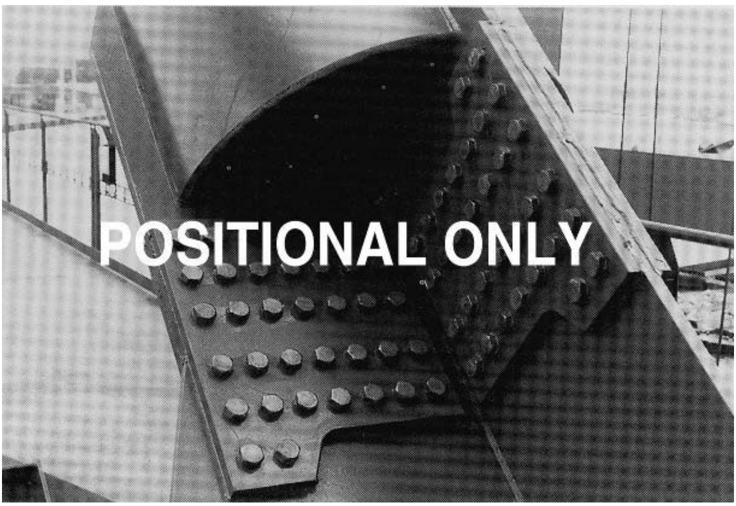


Fig. 7.5.8 Bolts in a back tie

7.6 Calculations on strength and on fatigue

In this section we follow primarily the rules of the FEM (see Section 5.2) as they give a clear and respected view on this subject. Fatigue in constructions, structures and mechanisms is dangerous; it can lead to severe breakdowns. Fatigue must be distinguished in structures, and fatigue in mechanisms or mechanical components. The calculation of the strength of a crane and of fatigue are two rather different items. In the strength calculation, the maximum loads and tensions in a certain area have to be calculated. The prescribed coefficients (see for example FEM standards as mentioned here) must be respected, as well as the allowed tensions. However when the same part of a construction has to bear 2 million or even 4 million times a heavy, fluctuating load, the allowed tensions are not the same, but far lower; especially when the construction details are taken into account. For calculation of fatigue the maximum tension and minimum tension in the construction detail have to be calculated.

It is clear that in a bulk unloader the maximum load is the loaded or overloaded grab, which runs to and fro over the boom and bridge girders every 45 seconds. In a container crane the container can be empty (weighing 2–3 tons), partly loaded or fully loaded.

A calculation of the load which is called the fatigue load shows quite something else, as the average weights of containers are rather low.

A – The fatigue load

This fatigue load can for instance be calculated as follows:

$$F_1 = \sqrt[3]{\Sigma\left[F^3 \cdot \frac{p\%}{100}\right]} \text{ tons}$$

where:

F_1 = the fatigue load in tons;
F = the load on the wire ropes in tons;
p percent = the percentage that the container, weighing F tons, is transported through the crane during a considerably long time.

Table 7.6.1 Calculating the fatigue load

Load under the spreader in tons	*Load 'on the ropes' F in tons*	*Percentage p %*	*Fatigue load $[F^3 \cdot p\,\%/100]$*
'empties'	20	20	1600
10	24	20	2765
20	34	12	4716
26	40	20	12 800
30	44	10	8518
36	50	5	6250
40	54	5	7873
50	64	4	10 486
60	74	4	16 209
			72 217

A terminal with a very high throughput measured the following averages:

Container ship to shore crane

- Max. load under the spreader: 60 tons
- Max. load 'on the ropes'
  (Container plus spreader): 60 + 14 = 74 tons.

$$F_1 = \sqrt[3]{\Sigma\left[F^3 \cdot \frac{p\%}{100}\right]} = \sqrt[3]{71\,217} = 41{,}45 \text{ tons}$$

A crane for this terminal should be calculated for strength with a load of 74 tons on the ropes and for fatigue with a fatigue load $F = 41{,}5$ tons, being 56 percent of the maximum load on the ropes.

B – Fatigue in structures

In order to calculate the fatigue in structures, we first have to apply the rules of the FEM 1.001, 3rd Edition, rev. 1998, Booklets 2 and 3. Only an excerpt is given:

Table T.2.1.2.2 Classes of utilization

Symbol	Total duration of use (number n_{max} of hoisting cycles)				
U0			n_{max}	$\leq$	16 000
U1	16 000	$<$	n_{max}	$\leq$	32 000
U2	32 000	$<$	n_{max}	$\leq$	63 000
U3	63 000	$<$	n_{max}	$\leq$	125 000
U4	125 000	$<$	n_{max}	$\leq$	250 000
U5	250 000	$<$	n_{max}	$\leq$	500 000
U6	500 000	$<$	n_{max}	$\leq$	1 000 000
U7	1 000 000	$<$	n_{max}	$\leq$	2 000 000
U8	2 000 000	$<$	n_{max}	$\leq$	4 000 000
U9	4 000 000	$<$	n_{max}		

By *duration of use* of a hoisting appliance is meant the number of hoisting cycles which the appliance performs. A hoisting cycle is the entire sequence of operations commencing when a load is hoisted and ending at the moment when the appliance is ready to hoist the next load.

The *total duration of use* is a computed duration of use, considered as a guide value, commencing when the appliance is put into service and ending when it is finally taken out of service.

On the basis of the total duration of use, we have ten classes of utilization, designated by the symbols U0, U1, . . . , U9. They are defined in Table T.2.1.2.2.

2.1.2.3 LOAD SPECTRUM

The load spectrum characterizes the total number of loads hoisted during the total duration of use (see 2.1.2.2) of an appliance. It is a distribution function (summed) $y = f(x)$, expressing the fraction x ($0 \le x \le 1$) of the total duration of use, during which the ratio of the hoisted load to the safe working load attains at least a given value y ($0 \le y \le 1$).

According to its load spectrum, a hoisting appliance is placed in one of the four spectrum classes Q1, Q2, Q3, Q4 defined in Table T.2.1.2.3.

Table T.2.1.2.3 Spectrum classes

Symbol	Spectrum factor k_p
Q1	$k_p \le 0.125$
Q2	$0.125 < k_p \le 0.250$
Q3	$0.250 < k_p \le 0.500$
Q4	$0.500 < k_p \le 1.000$

Table T.2.1.2.4 Appliance groups

Load spectrum class	Class of utilization									
	U0	U1	U2	U3	U4	U5	U6	U7	U8	U9
Q1	A1	A1	A1	A2	A3	A4	A5	A6	A7	A8
Q2	A1	A1	A2	A3	A4	A5	A6	A7	A8	A8
Q3	A1	A2	A3	A4	A5	A6	A7	A8	A8	A8
Q4	A2	A3	A4	A5	A6	A7	A8	A8	A8	A8

Table T.2.1.2.5 Guidance for group classification of appliances

Ref.	Type of appliance: Designation	Particulars concerning nature of use[(1)]	Appliance group (see 2.1.2.4)
1	Hand-operated appliances		A1–A2
2	Erection cranes		A1–A2
3	Erection and dismantling cranes for power stations, machine shops, etc.		A2–A4
4	Stocking and reclaiming transporters	Hook duty	A5
5	Stocking and reclaiming transporters	Grab or magnet	A6–A8
6	Workshop cranes		A3–A5
7	Overhead travelling cranes, pig-breaking cranes, scrapyard cranes	Grab or magnet	A6–A8
8	Ladle cranes		A6–A8
9	Soaking-pit cranes		A8
10	Stripper cranes, open-hearth furnace-charging cranes		A8
11	Forge cranes		A6–A8
12(a)	Bridge cranes for unloading, bridge cranes for containers	Hook or spreader duty	A5–A6
12(b)	Other bridge cranes (with crab and/or slewing jib crane)	Hook duty	A4

Table T.2.1.2.5 Continued

Ref.	Type of appliance Designation	Particulars concerning nature of use[(1)]	Appliance group (see 2.1.2.4)
13	Bridge cranes for unloading, bridge cranes (with crab and/or slewing jib crane)	Grab or magnet	A6–A8
14	Drydock cranes, shipyard jib cranes, jib cranes for dismantling	Hook duty	A3–A5
15	Dockside cranes (slewing, on gantry), floating cranes and pontoon derricks	Hook duty	A5–A6
16	Dockside cranes (slewing, on gantry), floating cranes and pontoon derricks	Grab or magnet	A6–A8
17	Floating cranes and pontoon derricks for very heavy loads (usually greater than 100 t)		A2–A3
18	Deck cranes	Hook duty	A3–A4
19	Deck cranes	Grab or magnet	A4–A5
20	Tower cranes for building		A3–A4
21	Derricks		A2–A3
22	Railway cranes allowed to run in train		A4

[(1)]Only a few typical cases of uses are shown, by way of guidance, in this column.

2.1.3.2 CLASSES OF UTILIZATION

By *duration of use* of a mechanism is meant the time during which the mechanism is actually in motion.

The *total duration of use* is a calculated duration of use, considered as a guide value, applying up to the time of replacement of the mechanism. It is expressed in terms of hours.

On the basis of this total duration of use, we have ten classes of utilization, T0, T1, T2, . . . , T9. They are defined in Table T.2.1.3.2.

Table T.2.1.3.2 Classes of utilization

Symbol	Total duration of use T (h)				
T0			T	≤	200
T1	200	<	T	≤	400
T2	400	<	T	≤	800
T3	800	<	T	≤	1600
T4	1600	<	T	≤	3200
T5	3200	<	T	≤	6300
T6	6300	<	T	≤	12 500
T7	12 500	<	T	≤	25 000
T8	25 000	<	T	≤	50 000
T9	50 000	<	T		

2.1.3.3 LOADING SPECTRUM

The loading spectrum characterizes the magnitude of the loads acting on a mechanism during its total duration of use. It is a distribution function (summed) $y = f(x)$, expressing the fraction x ($0 < x \leq$) of the total duration of use, during which the mechanism is subjected to a loading attaining at least a fraction y ($0 \leq y \leq 1$) of the maximum loading (see Fig. 2.1.2.3.1).

Table T.2.1.3.3 Spectrum classes

Symbol	Spectrum factor k_m
L1	$k_m \leq 0.125$
L2	$0.125 < k_m \leq 0.250$
L3	$0.250 < k_m \leq 0.500$
L4	$0.500 < k_m \leq 1.000$

2.1.3.4 GROUP CLASSIFICATION OF INDIVIDUAL MECHANISMS AS A WHOLE

On the basis of their class of utilization and their spectrum class, individual mechanisms as a whole are classified in one of the eight groups M1, M2, . . . , M8, defined in Table T.2.1.3.4.

Table T.2.1.3.4 Mechanism groups

Class of load spectrum	Class of utilization									
	T0	T1	T2	T3	T4	T5	T6	T7	T8	T9
L1	M1	M1	M1	M2	M3	M4	M5	M6	M7	M8
L2	M1	M1	M2	M3	M4	M5	M6	M7	M8	M8
L3	M1	M2	M3	M4	M5	M6	M7	M8	M8	M8
L4	M2	M3	M4	M5	M6	M7	M8	M8	M8	M8

2.1.3.5 GUIDE FOR GROUP CLASSIFICATION OF INDIVIDUAL MECHANISMS AS A WHOLE

Guidance for group classification of an individual mechanism as a whole is given in Table T.2.1.3.5.

Since appliances of the same type may be used in a wide variety of ways, the grouping directions in this table can only be taken as a model. In particular, where several groups are shown as appropriate to a mechanism of a given type, it is necessary to ascertain, on the basis of the mechanism's calculated total duration of use and loading spectrum, in which class of utilization (see 2.1.3.2) and spectrum (see 2.1.3.3) it has to be placed, and consequently in which group of mechanisms (see 2.1.3.4).

Table T.2.1.3.5 Guidance for group classification of a mechanism

Ref.	Types of appliance: Designation	Particulars concerning nature of use[1]	Type of mechanism: Hoisting	Slewing	Luffing	Traverse	Travel
1	Hand-operated appliances		M1	—	—	M1	M1
2	Erection cranes		M2–M3	M2–M3	M1–M2	M1–M2	M2–M3
3	Erection and dismantling cranes for power stations, machine shops, etc.		M2	—	—	M2	M2
4	Stocking and reclaiming transporters	Hook duty	M5–M6	M4	—	M4–M5	M5–M6
5	Stocking and reclaiming transporters	Grab or magnet	M7–M8	M6	—	M6–M7	M7–M8
6	Workshop cranes		M6	M4	—	M4	M5
7	Overhead travelling cranes, pig-breaking cranes, scrapyard cranes	Grab or magnet	M8	M6	—	M6–M7	M7–M8
8	Ladle cranes		M7–M8	—	—	M4–M5	M6–M7
9	Soaking-pit cranes		M8	M6	—	M7	M8
10	Stripper cranes, open-hearth furnace-charging cranes		M8	M6	—	M7	M8
11	Forge cranes		M8	—	—	M5	M6
12(a)	Bridge cranes for unloading, bridge cranes for containers	(a) Hook or spreader duty	M6–M7	M5–M6	M3–M4	M6–M7	M4–M5
12(b)	Other bridge cranes (with crab and/or slewing jib crane)	(b) Hook duty	M4–M5	M4–M5	—	M4–M5	M4–M5
13	Bridge cranes for unloading, bridge cranes (with crab and/or slewing jib crane)	Grab or magnet	M8	M5–M6	M3–M4	M7–M8	M4–M5
14	Drydock cranes, shipyard jib cranes, jib cranes for dismantling	Hook duty	M5–M6	M4–M5	M4–M5	M4–M5	M5–M6
15	Dockside cranes (slewing on gantry, etc.), floating cranes and pontoon derricks	Hook duty	M6–M7	M5–M6	M5–M6	—	M3–M4
16	Dockside cranes (slewing, on gantry, etc.), floating cranes and pontoon derricks	Grab or magnet	M7–M8	M6–M7	M6–M7	—	M4–M5
17	Floating cranes and pontoon derricks for very heavy loads (usually greater than 100 t)		M3–M4	M3–M4	M3–M4	—	—
18	Deck cranes	Hook duty	M4	M3–M4	M3–M4	M2	M3
19	Deck cranes	Grab or magnet	M5–M6	M3–M4	M3–M4	M4–M5	M3–M4
20	Tower cranes for building		M4	M5	M4	M3	M3
21	Derricks		M2–M3	M1–M2	M1–M2	—	—
22	Railway cranes allowed to run in train		M3–M4	M2–M3	M2–M3	—	—

[1] Only a few typical cases of use are shown, by way of guidance, in this column.

2.1.4 CLASSIFICATION OF COMPONENTS

2.1.4.1 CLASSIFICATION SYSTEM

Components, both structural and mechanical, are classified in eight groups, designated respectively by the symbols E1, E2, . . . , E8, on the basis of eleven classes of utilization and four classes of stress spectrum.

2.1.4.2 CLASSES OF UTILIZATION

By duration of use of a component is meant the number of stress cycles to which the component is subjected.

Table T.2.1.4.2 Classes of utilization

Symbol	Total duration of use (number n of stress cycles)				
B0			n	≤	16 000
B1	16 000	<	n	≤	32 000
B2	32 000	<	n	≤	63 000
B3	63 000	<	n	≤	125 000
B4	125 000	<	n	≤	250 000
B5	250 000	<	n	≤	500 000
B6	500 000	<	n	≤	1 000 000
B7	1 000 000	<	n	≤	2 000 000
B8	2 000 000	<	n	≤	4 000 000
B9	4 000 000	<	n	≤	8 000 000
B10	8 000 000	<	n		

2.1.4.3 STRESS SPECTRUM

The stress spectrum characterizes the magnitude of the load acting on the component during its total duration of use.

Depending on its stress spectrum, a component is placed in one of the spectrum classes P1, P2, P3, P4, defined in Table T.2.1.4.3.[1]

Table T.2.1.4.3 Spectrum classes

Symbol	Spectrum factor k_{sp}				
P1			k_{sp}	≤	0.125
P2	0.125	<	k_{sp}	≤	0.250
P3	0.250	<	k_{sp}	≤	0.500
P4	0.500	<	k_{sp}	≤	1.000

[1] There are components, both structural and mechanical, such as spring-loaded components, which are subjected to loading that is quite or almost independent of the working load. Special care shall be taken in classifying such components. In most cases $k_{sp} = 1$ and they belong to class P4.

2.1.4.4 GROUP CLASSIFICATION OF COMPONENTS

On the basis of their class of utilization and their stress spectrum class, components are classified in one of the eight groups E1, E2, . . . , E8, defined in Table T.2.1.4.4.

Table T.2.1.4.4 Component groups

Stress spectrum class	Class of utilization										
	B0	B1	B2	B3	B4	B5	B6	B7	B8	B9	B10
P1	E1	E1	E1	E1	E2	E3	E4	E5	E6	E7	E8
P2	E1	E1	E1	E2	E3	E4	E5	E6	E7	E8	E8
P3	E1	E1	E2	E3	E4	E5	E6	E7	E8	E8	E8
P4	E1	E2	E2	E4	E5	E6	E7	E8	E8	E8	E8

2.2.2.1 LOADS DUE TO HOISTING OF THE WORKING LOAD

Account shall be taken of the oscillations caused when lifting the load by multiplying the loads due to the working load by a factor called the 'dynamic coefficient Ψ'.

2.2.2.1.1 VALUES OF THE DYNAMIC COEFFICIENT Ψ

The value of the dynamic coefficient Ψ to be applied to the load arising from the working load is given by the expression:

$$\Psi = 1 + \xi v_L$$

Where v_L is the hoisting speed in m/s and ξ an experimentally determined coefficient.[1]

The following values shall be adopted:

$\xi = 0{,}6$ for overhead travelling cranes and bridge cranes;
$\xi = 0{,}3$ for jib cranes.

The maximum figure to be taken for the hoisting speed when applying this formula is 1 m/s. For higher speeds, the dynamic coefficient Ψ is not further increased.

The value to be applied for the coefficient Ψ in the calculations shall in no case be less than 1,15.

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(1) The figure given for this coefficient ξ is the result of a large number of measurements made on different types of appliances.

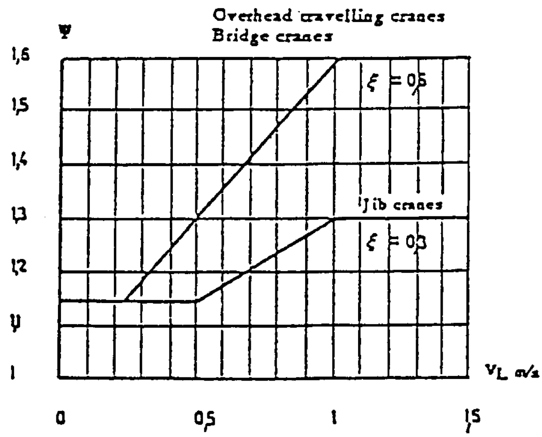


Fig. 2.2.2.1.1 Values of dynamic coefficient Ψ

The values of Ψ are given in the curves of Fig. 2.2.2.1.1 in terms of hoisting speeds v_L.

2.3 CASES OF LOADING

Three different cases of loading are to be considered for the purpose of the calculations:

– the working case without wind,
– the working case with limiting working wind,
– the case of exceptional loadings.

Having determined the various loads in accordance with Section 2.2, account is taken of a certain probability of exceeding the calculated stress, which results from imperfect methods of calculation and unforseen contingencies, by applying an amplifying coefficient γ_C, which varies according to the group classification of the appliance.

The values of this coefficient γ_C are indicated in clause 2.3.4.

2.3.1 CASE I: APPLIANCE WORKING WITHOUT WIND

The following shall be taken into consideration: the static loads due to the dead weight S_G, the loads due to the working load S_L multiplied by the dynamic coefficient Ψ, and the two most unfavourable horizontal effects S_H among those defined in clause 2.2.3, excluding buffer forces.

All these loads must then be multiplied by the amplifying coefficient γ_C specified in clause 2.3.4, viz:

$$\gamma_C(S_G + \Psi S_L + S_H)$$

In cases where travel motion takes place only for positioning the appliance and is not normally used for moving loads the effect of this motion shall not be combined with another horizontal motion. This is the case for example with a dockside crane which, once it has been positioned, handles a series of loads at a fixed point.

2.3.2 CASE II: APPLIANCE WORKING WITH WIND

The loads of case I are taken to which are added the effects of the limiting working wind S_W defined under 2.2.4.1.2.1. (Table T.2.2.4.1.2.1) and, where, applicable the load due to temperature variation, viz:

$$\gamma_C(S_G + \Psi S_L + S_H) + S_W$$

Note – The dynamic effects of acceleration and retardation do not have the same values in case II as in case I, for when a wind is blowing the accelerating or braking times are not the same as when still conditions prevail.

2.3.3 CASE III: APPLIANCE SUBJECTED TO EXCEPTIONAL LOADINGS

Exceptional loadings occur in the following cases:

– appliance out of service with maximum wind,
– appliance working and subjected to a buffer effect,
– appliance undergoing the tests indicated in booklet 8.

The highest of the following combinations shall be considered:

(a) The loads S_G due to the dead weight, plus the load $S_{W\,max}$ due to the maximum wind as mentioned under clause 2.2.4.1.2.2 (including the reactions of the anchorages);
(b) the loads S_G due to the dead weight and S_L due to the working load plus the greatest buffer effect S_T as envisaged in clause 2.2.3.4;
(c) the loads S_G due to the dead weight plus the highest of the two loads $\Psi\rho_1 S_L$ and $\rho_2 S_L$; ρ_1 and ρ_2 being the coefficients by which the safe working load is multiplied for the dynamic test (ρ_1) and for the static test (ρ_2) as in clauses 8.1.1 and 8.1.2.

These three cases are expressed by the formulae:

(a) $S_G + S_{W\,max}$
(b) $S_G + S_L + S_T$ (1)
(c) $S_G + \Psi\rho_1 SL$ or $S_G + \rho_2 S_L$

Note 1 – It should be noted that the checks under (c) are only to be made in cases where the working load, when assumed to act alone, produces stresses opposed in direction to those caused by the dead weight up to the point at which the static test load does not exceed 1,5 times the safe working load.

Note 2 – When using decelerating devices in advance of buffer impact under the conditions mentioned in clause 2.2.3.4.1, S_T will be taken to be the highest load resulting either from the retardation previously caused by the decelerating device or from that finally caused by the buffer.

2.3.4 CHOOSING THE AMPLIFYING COEFFICIENT γ_C

The value of the amplifying coefficient γ_C depends upon the group classification of the appliance.

Table T.2.3.4 Values of amplifying coefficient γ_C

Appliance group	A1	A2	A3	A4	A5	A6	A7	A8
γ_C	1.00	1.02	1.05	1.08	1.11	1.14	1.17	1.20

Article 2.5 of FEM: Loads entering into the design of mechanisms and Article 2.6 of FEM: Cases of loading of mechanisms have not been reproduced in the book *Cranes*.

3.1.3 QUALITY OF STEELS

The quality of steels in these design rules is the property of steel to exhibit a ductile behaviour at determined temperatures.

The steels are divided into four quality groups. The group in which the steel is classified, is obtained from its notch ductility in a given test and temperature.

Table T.3.1.3 comprises the notch ductility values and test temperatures for the four quality groups.

The indicated notch ductilities are minimum values, being the mean values from three tests, where no value must be below 20 Nm/cm^2.

The notch ductility is to be determined in accordance with V-notch impact tests to ISO R 148 and Euronorm 45–63.

Steels of different quality groups can be welded together.

T_C is the test temperature for the V-notch impact test.

T is the temperature at the place of erection of the crane.

T_C and T are not directly comparable as the V-notch impact test imposes a more unfavourable condition than the loading on the crane in or out of service.

Table T.3.1.3 Quality groups

Quality group	Notch ductility measured in ISO sharp notch test ISO R 148 in Nm/cm^2	Test temperature T_C (°C)	Steels, corresponding to the quality group Designation of steels	Standard
1	—	—	Fe 360 – A Fe 430 – A	Euronorm 25
			St 37 – 2 St 44 – 2	DIN 17100
			E 24 – 1	NF A 35–501
			43 A 50 B*	BS 4360 1972
2	35	+ 20°	Fe 360 – B Fe 430 – B Fe 510 – B	Euronorm 25
			R St 37–2 St 44–2	DIN 17100
			E 24 (A37) – 2 E 26 (A42) – 2 E 36 (A52) – 2	NF A 35–501
			40 B 43 B*	BS 4360 1972
3	35	± 0°	Fe 360 – C Fe 430 – C Fe 510 – C	Euronorm 25
			St 37 – 3U St 44 – 3U St 52 – 3U	DIN 17100
			E 24 (A37) – 3 E 26 (A42) – 3 E 36 (A52) – 3	NF A 35–501
			40 C 43 C* 50 C 55 C*	BS 4360 1972
4	35	–20°	Fe 360 – D Fe 410 – D Fe 510 – D	Euronorm 25
			St 37 – 3N St 44 – 3N St 52 – 3N	DIN 17100
			E 24 (A37) – 4 E 26 (A42) – 4 E 36 (A52) – 4	NF A 35–501
			40 D 43 D* 50 D 55 E*	BS 4360 1972

*The test requirements of steels to BS 4360 do not in all cases agree with the Euronorm and other national standards, and the guaranteed impact test properties for steels to BS 4360 may be different to other steels in the same quality group. Impact test properties are stated in BS 4360 and where the requirements are different from those guaranteed in BS 4360, agreement must be obtained from the steel suppliers.

3.1.4 SPECIAL RULES

In addition to the above provisions for the choice of the steel quality, the following rules are to be observed:

1. Non-killed steels of group 1 shall be used for load carrying structures only in case of rolled sections and tubes not exceeding 6 mm thickness.
2. Members of more than 50 mm thickness, shall not be used for welded load carrying structures unless the manufacturer has a comprehensive experience in the welding of thick plates. The steel quality and its testing has in this case to be determined by specialists.
3. If parts are cold bent with a radius/plate thickness ratio < 10 the steel quality has to be suitable for folding or cold flanging.

3.2 CHECKING WITH RESPECT TO THE ELASTIC LIMIT

For this check, a distinction is made between the actual members of the structure and the riveted, bolted or welded joints.

3.2.1 STRUCTURAL MEMBERS OTHER THAN JOINTS

3.2.1.1 MEMBERS SUBJECTED TO SIMPLE TENSION OR COMPRESSION

(1) Case of steels for which the ratio between the elastic limit σ_E and the ultimate tensile strength σ_R is $<0,7$.

The computed stress σ must not exceed the maximum permissible stress σ_a obtained by dividing the elastic limit stress σ_E by the coefficient ν_E which depends upon the case of loading as defined under Section 2.3.

The values of ν_E and the permissible stresses are:

	Case I	Case II	Case III
Values of ν_E	1,5	1,33	1,1
Permissible stresses σ_a	$\sigma_E/1{,}5$	$\sigma_E/1{,}3$	$\sigma_E/1{,}1$

For carbon steels of current manufacture A.37 – A.42 – A.52 (also called E.24 – E.26 – E.36 or Fe 360 – Fe 510) the critical stress σ_E is conventionally taken as that which corresponds to an elongation of 0,2 percent.

Table T.3.2.1.1 Values of σ_E and σ_a for steels A.37 – A.42 – A.52

Steels	Elastic limit σ_E N/mm^2	Maximum permissible stresses: σ_a Case I N/mm^2	Case II N/mm^2	Case III N/mm^2
E.24 (A.357, Fe 360)	240	160	180	215
E.26 (A.42)	260	175	195	240
E.36 (A.52, Fe 510)	360	240	270	325

3.2.1.2 MEMBERS SUBJECTED TO SHEAR

The permissible stress in shear τ_a has the following value:

$$\tau_a = \frac{\sigma_a}{\sqrt{3}}$$

σ_a being the permissible tensile stress.

3.2.1.3 MEMBERS SUBJECTED TO COMBINED LOADS – EQUIVALENT STRESS

σ_x, σ_y and τ_{xy} being respectively the two normal stresses and the shear stress at a given point, a check shall be made:

1. That each of the two stresses σ_x and σ_y is less than σ_a and that τ_{xy} is less than τ_a,
2. that the equivalent stress σ_{cp} is less than σ_a, i.e.:

$$\sigma_{cp} = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau_{xy}^2} \leq \sigma_a$$

When using this formula, a simple method is to take the maximum values σ_x, σ_y and τ_{xy}. But, in fact, such a calculation leads to too great an equivalent stress if it is impossible for the maximum values of each of the three stresses to occur simultaneously.

Nevertheless, the simple calculation method, being conservative, is always acceptable.

If it is desired to calculate more precisely, it is necessary to determine the most unfavourable practical combination that may occur. Three checks must then be made by calculating successively the equivalent stress resulting from the three following combinations:

$\sigma_{x\,max}$ and the corresponding stresses σ_y and τ_{xy}

$\sigma_{y\,max}$ and the corresponding stresses σ_x and τ_{xy}

$\tau_{xy\,max}$ and the corresponding stresses σ_x and σ_y

Note: It should be noted that when two out of the three stresses are approximately of the same value, and greater than half the permissible stress, the most unfavourable combination of the three values may occur in different loading cases from those corresponding to the maximum of each of the three stresses.

Special case:

– Tension (or compression) combined with shear

The following formula should be checked:

$$\sqrt{\sigma^2 + 3\tau^2} \leq \sigma_a$$

3.2.2.3 WELDED JOINTS

In welded joints, it is assumed that the deposited metal has at least as good characteristics as the parent metal.

It must be verified that the stresses developed, in the cases of longitudinal tension and compression, do not exceed the permissible stresses σ_a given in clause 3.2.1.1.

For shear in the welds, the permissible stress τ_a is given by:

$$\tau_a = \frac{\sigma_q}{\sqrt{2}}$$

Table T.3.2.2.3 Maximum permissible equivalent stresses in welds (N/mm^2) steels A.37 (Fe 360) – A.42 – A.52 (Fe 510)

	A.37			A.42			A.52		
Types of loading	Case I	Case II	Case III	Case I	Case II	Case III	Case I	Case II	Case III
Longitudinal equivalent stresses for all types of welds	160	180	215	175	195	240	240	270	325
Transverse tensile stresses									
1. Butt-welds and special quality K-welds	160	180	215	175	195	240	240	270	325
2. Ordinary quality K-welds	140	158	185	153	170	210	210	236	285
3. Fillet welds	113	127	152	124	138	170	170	191	230
Transverse compressive stresses									
1. Butt-welds and K-welds	160	180	215	175	195	240	240	270	325
2. Fillet welds	130	146	175	142	158	195	195	220	265
Shear All types of welds	113	127	152	124	138	170	170	191	230

However, for certain types of loading, particularly transverse stresses in the welds, the maximum permissible equivalent stress is reduced.

Table T.3.2.2.3 summarizes the values not to be exceeded, for certain steels, according to the type of loading.

Appendix A-3.2.2.3 gives some additional information on welded joints.

3.6 CHECKING MEMBERS SUBJECTED TO FATIGUE

Danger of fatigue occurs when a member is subjected to varying and repeated loads.

Fatigue strength is calculated by considering the following parameters:

1 – the conventional number of cycles and the stress spectrum to which the member is subjected;
2 – the material used and the notch effect at the point being considered;
3 – the extreme maximum stress σ_{max} which can occur in the member;
4 – the ratio κ between the values of the extreme stresses.

3.6.1 CONVENTIONAL NUMBERS OF CYCLES AND STRESS SPECTRUM

The number of cycles of variations of loading and the spectrum of stresses to be taken into consideration are discussed in clause 2.1.2.2 and in clause 2.1.2.3.

These two parameters are taken into account when considering solely the group in which the member is classified in accordance with clause 2.1.4.

3.6.2 MATERIAL USED AND NOTCH EFFECT

The fatigue strength of a member depends upon the quality of the material used and upon the shape and the method of making the joints. The shapes of the parts joined and the means of doing it have the effect of producing stress concentrations (or notch effects) which considerably reduce the fatigue strength of the member.

Appendix A-3.6 gives a classification of various joints according to their degree of stress concentration (or notch effect).

3.6.3 DETERMINATION OF THE MAXIMUM STRESS σ_{max}

The maximum stress, σ_{max}, is the highest stress in absolute value (i.e. it may be tension or compression) which occurs in the member in loading case I referred to in clause 2.3.1 without the application of the amplifying coefficient γ_c.

When checking members in compression for fatigue the crippling coefficient, ω, given in clause 3.3 should not be applied.

3.6.4 THE RATIO κ BETWEEN THE EXTREME STRESSES

This ratio is determined by calculating the extreme values of the stresses to which the component is subjected under case I loadings.

The ratio may vary depending upon the operating cycles but it errs on the safe side to determine this ratio κ by taking the two extreme values which can occur during possible operations under case I loadings.

If σ_{max} and σ_{min} are the algebraic values of these extreme stresses, σ_{max} being the extreme stress having the higher absolute value, the ratio κ may be written:

$$\kappa = \frac{\sigma_{min}}{\sigma_{max}} \quad \text{or} \quad \frac{\tau_{min}}{\tau_{max}} \text{ in the case of shear}$$

This ratio, which varies from +1 to −1, is positive if the extreme stresses are both of the same sense (fluctuating stresses) and negative when the extreme stresses are of opposite sense (alternating stresses).

3.6.5 CHECKING MEMBERS SUBJECTED TO FATIGUE

Using the parameters defined in clauses 3.6.1 to 3.6.4 the adequacy of the structural members and of the joints subjected to fatigue is ensured by checking that the stress σ_{max}, as defined in clause 3.6.3 is not greater than the permissible stress for fatigue of the members under consideration.

This permissible stress for fatigue is derived from the critical stress, defined as being the stress which, on the basis of tests made with test pieces, corresponds to a 90 percent probability of survival to which a coefficient of safety of 4/3 is applied thus:

σ_a for fatigue $= 0{,}75\sigma$ at 90 percent survival.

The determination of these permissible stresses having regard to all these considerations is a complex problem and it is generally advisable to refer to specialized books on the subject.

Appendix A-3.6 gives practical indications, based on the results of research in this field, on the determination of permissible stresses for A.37 – A.42 and A.52 steels, according to the various groups in which the components are classified, and the notch effects of the main types of joints used in the manufacture of hoisting appliances.

APPENDIX A-3.6

CHECKING STRUCTURAL MEMBERS SUBJECT TO FATIGUE

It must be remembered that fatigue is one of the causes of failure envisaged in clause 3.6 and therefore checking for fatigue is additional to checking in relation to the elastic limit or permissible crippling or buckling.

If the permissible stresses for fatigue, as determined hereunder, are higher than those allowed for other conditions then this merely indicates that the dimensions of the components are not determined by considerations of fatigue.

Clause 3.6 enumerates the parameters which must be considered when checking structural components for fatigue.

The purpose of this appendix is firstly to classify the various joints according to their notch effect, as defined in clause 3.6.2 and, then, to determine for these various notch effects and for each classification group of the component as defined in clause 2.1.4 the permissible stresses for fatigue as a function of the coefficient κ defined in clause 3.6.4.

These permissible fatigue stresses were determined as a result of tests carried out by the F.E.M. on test pieces having different notch effects and submitted to various loading spectra. They were determined on the basis of the stress values which, in the tests, assured 90 percent survival including a factor of a safety of 4/3.

In practice, a structure consists of members which are welded, riveted or bolted together and experience shows that the behaviour of a member differs greatly from one point to another; the immediate proximity of a joint invariably constitutes a weakness that will be vulnerable to a varying extent according to the method of assembly used.

An examination is therefore made in the first sections, of the effect of fatigue on structural members both away from any joint and in immediate proximity to the usual types of joint.

The second section examines the resistance to fatigue of the means of assembly themselves, i.e. weld seams, rivets and bolts.

1 VERIFICATION OF STRUCTURAL MEMBERS

The starting point is the fatigue strength of the continuous metal away from any joint and, in general, away from any point at which a stress concentration, and hence a lessening of the fatigue strength, may occur.

In order to make allowance for the reduction in strength near joints, as a result of the presence of holes or welds producing changes of section, the notch effects in the vicinity of these joints, which characterize the effects of the stress concentrations caused by the presence of discontinuities in the metal, are examined.

These notch effects bring about a reduction of the permissible stresses, the extent of which depends upon the type of discontinuity encountered, i.e. upon the method of assembly used.

In order to classify the importance of these notch effects, the various forms of joint construction are divided into categories as follows:

Unwelded parts

These members present three cases of construction.

Case W_0 concerns the material itself without notch effect.
Cases W_1 and W_2 concern perforated members (see Table T.A.3.6 (1)).

Welded parts

These joints are arranged in order of the severity of the notch effect increasing from K_0 to K_4, corresponding to structural parts located close to the weld fillets.

Table T.A.3.6 (1) gives some indications as to the quality of the welding and a classification of the welding and of the various joints that are most often used in the construction of lifting appliances.

Determination of the permissible stresses for fatigue

Tensile and compressive loads

The basis values which have been used to determine the permissible stresses in tension and compression are those resulting from application of a constant alternating stress $\pm\sigma_w$ ($\kappa = -1$) giving a survival rate of 90 percent in the tests, to which a factor of safety of 4/3 has been applied.

To take account of the number of cycles and of the stress spectrum, the σ_w values have been set for each classification group of the member the latter taking account of these two parameters.

For unwelded parts, the values σ_w are identical for steel St 37, and St 44. They are higher for St 52.

For welded parts, the σ_w values are identical for the three types of steel.

Table T.A.3.6.1 Values of σ_w depending on the component group and construction case (N/mm^2)

Component group	Unwelded components Construction cases						Welded components Construction cases (Steels St 37 to St 52, Fe 360 to Fe 510)				
	W_0		W_1		W_2						
	Fe 360 St 37 St 44	St 52 Fe 510	Fe 360 St 37 St 44	St 52 Fe 510	Fe 360 St 37 St 44	St 52 Fe 510	K_0	K_1	K_2	K_3	K_4
E1	249,1	298,0	211,7	253,3	174,4	208,6	(361,9)	(323,1)	(271,4)	193,9	116,3
E2	224,4	261,7	190,7	222,4	157,1	183,2	(293,8)	262,3	220,3	157,4	94,4
E3	202,2	229,8	171,8	195,3	141,5	160,8	238,4	212,9	178,8	127,7	76,6
E4	182,1	210,8	154,8	171,5	127,5	141,2	193,5	172,8	145,1	103,7	62,2
E5	164,1	177,2	139,5	150,6	114,9	124,0	157,1	140,3	117,8	84,2	50,5
E6	147,8	155,6	125,7	132,3	103,5	108,9	127,5	113,8	95,6	68,3	41,0
E7	133,2	136,6	113,2	116,2	93,2	95,7	103,5	92,4	77,6	55,4	33,3
E8	120,0	120,0	102,0	102,0	84,0	84,0	84,0	75,0	63,0	45,0	27,0

The values in brackets are greater than 0,75 times the breaking stress and are only theoretical values (see note 2 at the end of this clause).

The following formulae give for all values of κ the permissible stresses for fatigue:

(a) $\kappa \leq 0$

– for tension: $$\sigma_t = \sigma_w \frac{5}{3 - 2\kappa} \quad (1)$$

– for compression: $$\sigma_c = \sigma_w \frac{2}{1 - \kappa} \quad (2)$$

σ_w is given in the table above.

(b) $\kappa > 0$

– for tension: $$\sigma_t = \frac{\sigma_0}{1 - \left(1 - \dfrac{\sigma_0}{\sigma_{+1}}\right)\kappa} \quad (3)$$

– for compression: $$\sigma_c = 1{,}2\sigma_t \quad (4)$$

where σ_0 = tensile stress for $\kappa = 0$ is given by the formula (1) that is:

$$\sigma_0 = 1{,}66\sigma_w$$

σ_{+1} = tensile stress for $\kappa = +1$ that is the ultimate strength σ_R divided by the coefficient of safety 4/3:

$$\sigma_{+1} = 0{,}75\sigma_R$$

σ_t is limited in every case to $0{,}75\sigma_R$.

By way of illustration, Fig. A.3.6.1 shows curves giving the permissible stress as a function of the ratio κ for the following cases:

- steel A.52;
- predominant tensile stress;
- group E6;
- construction cases W_0, W_1, W_2 for unwelded components and cases of construction for joints K_0 to K_4.

The permissible stresses have been limited to 240 N/mm^2, i.e. to the permissible stress adopted for checking for ultimate strength.

Table T.A.3.6 (1) Classification of cases of construction for joints

Joints may be riveted, bolted or welded.

The types of weld most commonly used for hoisting appliances are butt welds, double bevel welds (K welds) and fillet welds, of ordinary quality (O.Q.) or special quality (S.Q.) as specified below.

Weld testing is also stipulated for certain types of joint.

A *Weld qualities*

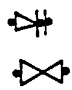
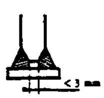
Type of weld	Weld quality	Execution of weld	Symbol(1)	Weld testing	Symbol
Full depth butt weld	Special quality (S.Q.)	Root of weld scraped (or trimmed) before making sealing run. No end craters. Weld ground flush with plate parallel to direction of forces		Check (e.g. with X-rays) over 100% of seam length	P 100
	Ordinary quality (O.Q.)	Root of weld scraped (or trimmed) before making sealing run. No end craters		If the calculated stress >80% times the permissible stress	P 100
				Otherwise random check over at least 10% of seam length	P 10
K-weld in angle formed by two parts with bevel on one of the parts to be joined at location of seam	Special quality (S.Q.)	Root of weld scraped (or trimmed) before making weld on other side. Weld edges without under-cutting and ground if necessary. Full penetration welds		Check that for tensile loads the plate perpendicular to the direction of the forces is free from lamination	D
	Ordinary quality (O.Q.)	Width clear of weld penetration between the two welds <3 mm 			
Fillet welds in the angle formed by two parts	Special quality (S.Q.)	Welded edges without undercutting and ground if necessary		Check that for tensile loads the plate perpendicular to the direction of the forces is free from lamination	D
	Ordinary quality (O.Q.)				

Table T.A.3.6 (1) – continued

B – *Cases of construction for joints*

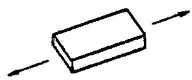
In the tables below the various cases of means of assembly are classified in terms of the magnitude of the notch effect they produce.

It should be noted that, with a given weld, the notch effect differs according to the type of loading to which the joint is subjected.

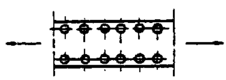
For example, a fillet-welded joint is classified under case K_0 for longitudinal tension or compression loads (0,31) or longitudinal shear (0,51), and under cases K_3 or K_4 for transverse tension or compression loads (3,2 or 4,4).

1 *Non welded parts*

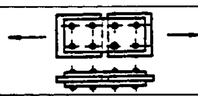
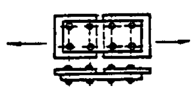
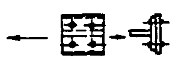
Case W_0

Reference	Description	Figure	Symbol
W_0	Parent metal, homogeneous surface. Part without joints or breaks in continuity (solid bars) and without notch effects unless the latter can be calculated.		

Case W_1

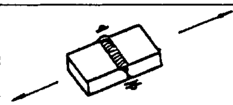
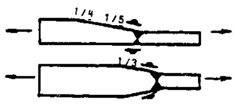
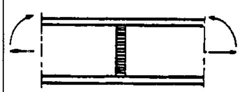
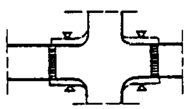
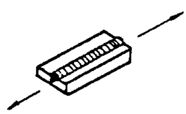
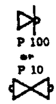
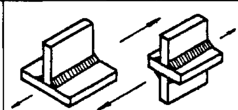
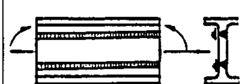
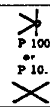
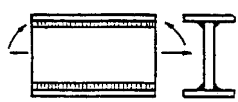
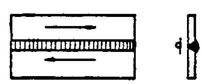
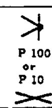
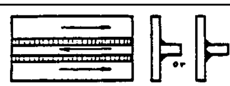
Reference	Description	Figure	Symbol
W_1	Parts drilled. Parts drilled for riveting or bolting with rivets and bolts loaded up to 20% of permissible values. Parts drilled for joints using high strength bolts (Cl 3.2.2.2.2.3) loaded up to 100% of permissible values (Cl 3.2.2.2.2.2)		

Case W_2

Reference	Description	Figure	Symbol
$W_{2,1}$	Parts drilled for riveting or bolting in which the rivets or bolts are loaded in multiple shear		
$W_{2,2}$	Parts drilled for riveting or bolting, in which the rivets or bolts are loaded in single shear (allowing for eccentric loads), the parts being unsupported		
$W_{2,3}$	Parts drilled for assembling by means of rivets or bolts loaded in single shear, the parts being supported or guided		

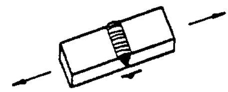
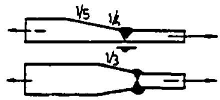
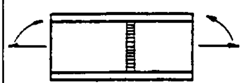
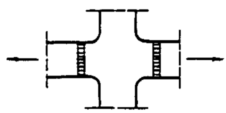
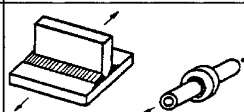
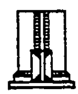
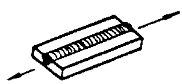
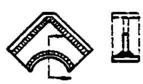
2 *Welded parts*

Case K_0 Slight stress concentration

Reference	Description	Figure	Symbol[1]
0,1	Parts butt-welded (S.Q.) at right angles to direction of forces		
0,11	Parts of different thickness butt-welded (S.Q.) at right angles to direction of forces. Asymmetrical slope: 1/4 to 1/5; Symmetrical slope: 1/3		
0,12	Butt weld (S.Q.) in transverse joint of web plate		
0,13	Gusset secured by butt-welding (S.Q.) at right angles to the direction of the forces		
0,3	Parts joined by butt-welding (O.Q.) parallel to the direction of the forces		
0,31	Parts joined by fillet welds (O.Q.) parallel to the direction of the forces (longitudinal to the joined parts)		
0,32	Butt weld (O.Q.) between section forming flange and web of a beam		
0,33	K- or fillet weld (O.Q.) between flange and web of a beam calculated for the equivalent stress for combined forces (Cl 3.2.1.3)		
0,5	Butt weld (O.Q.) in the case of longitudinal shear		
0,51	K-weld (O.Q.) or fillet weld (O.Q.) in the case of longitudinal shear		

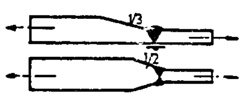
[1] It is forecast that the symbols shall be adapted to the ISO standard 2553 at the next edition of the Design Rules, when the addition of this standard will be definitively adopted.

Case K_1 Moderate stress concentration

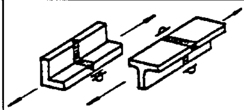
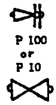
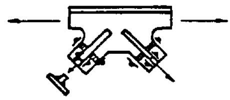
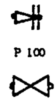
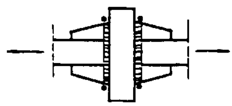
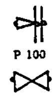
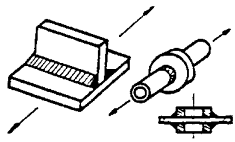
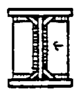
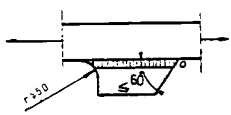
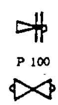
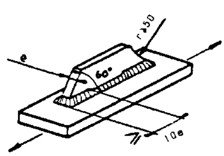
Reference	Description	Figure	Symbol[1]
1,1	Parts joined by butt welding (O.Q.) at right angles to the direction of the forces		P 100 or P 10
1,11	Parts of different thickness butt welded (O.Q.) at right angles to the direction of the forces. Asymmetrical slope: 1 in 4 to 1 in 5 (or symmetrical slopes: 1 in 3)		P 100 or P 10
1,12	Butt weld (O.Q.) executed for transverse joint of web plate		P 100 or P 10
1,13	Gusset joined by butt welding (O.Q.) at right angles to the direction of the forces		P 100 or P 10
1,2	Continuous main member to which are joined by continuous K-welds (S.Q.) parts at right angles to the direction of forces		
1,21	Web plate to which stiffeners are joined at right angles to the direction of the forces by means of fillet welds (S.Q.) which extend round the corners of the web stiffeners		
1,3	Parts joined by butt welding parallel to the direction of the forces (without checking the welding		
1,31	K-weld (S.Q.) between curved flange and web		

[1] It is forecast that the symbols shall be adapted to the ISO standard 2553 at the next edition of the Design Rules, when the addition of this standard will be definitively adopted.

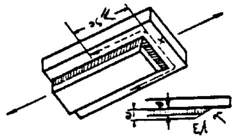
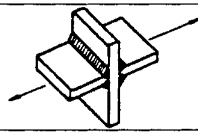
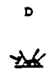
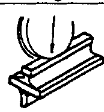
Case K_2 Medium stress concentration

Reference	Description	Figure	Symbol[1]
2,1	Parts of different thickness butt welded (O.Q.) at right angles to the direction of the forces. Asymmetrical slope: 1 in 3 (or symmetrical slopes: 1 in 2)		

Case K_2 Medium stress concentration (continued)

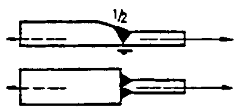
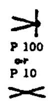
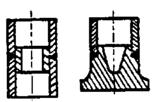
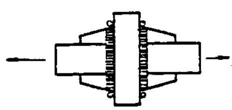
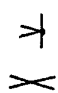
Reference	Description	Figure	Symbol[1]
2,11	Sections joined by butt welds (S.Q.) at right angles to the direction of the forces		
2,12	Section joined to a gusset by a butt weld (S.Q.) at right angles to the direction of the forces		
2,13	Butt weld (S.Q.) at right angles to the direction of the forces, made at intersection of flats, with welded auxiliary gussets. The ends of the welds are ground, avoiding notches		
2,2	Continuous main member to which transverse diaphragms, web stiffeners, rings or hubs are fillet welded (S.Q.) at right angles to the direction of the forces		
2,21	Web in which fillet welds (S.Q.) are used to secure transverse web stiffeners with cut corners, the welds not extending round the corners		
2,22	Transverse diaphragm secured by fillet welds (S.Q.) with cut corners, in which the welds do not extend round the corners		
2,3	Continuous main member to the edges of which are butt welded (S.Q.) parts parallel to the direction of the forces. These parts terminate in bevels or radii. The ends of the welds are ground avoiding notches		
2,31	Continuous main member to which are welded parts parallel to the direction of the forces. These parts terminate in bevels or radii. Valid where the ends of the welds are K-welds (S.Q.) over a length equal to ten times the thickness provided that the ends of the welds are ground avoiding notches		

Case K_2 Medium stress concentration (continued)

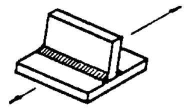
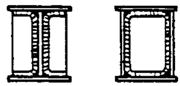
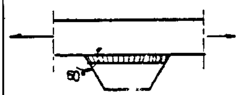
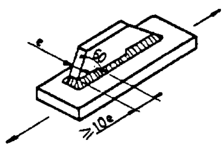
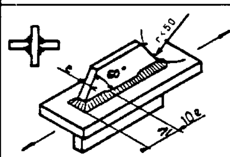
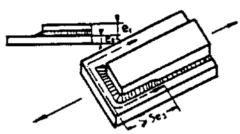
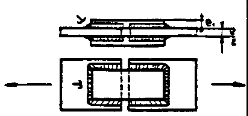
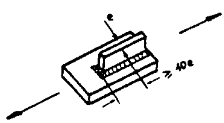
Reference	Description	Figure	Symbol[1]
2,33	Continuous member to which a flat (1 in 3 bevel) is joined by a fillet weld (S.Q.), the fillet weld being executed in the X area, with a = 0,5 e		
2,34	K-weld (O.Q.) made between curved flange and web		
2,4	Cruciform joint made with K-welds (S.Q.) perpendicular to the direction of the forces		
2,41	K-weld (S.Q.) between flange and web in the case of load concentrated in the plane of the web at right angles to the weld		
2,5	K-weld (S.Q.) joining parts stressed in bending or shear		

[1] It is forecast that the symbols shall be adapted to the ISO standard 2553 at the next edition of the Design Rules, when the addition of this standard will be definitively adopted.

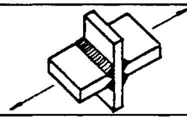
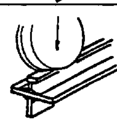
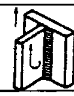
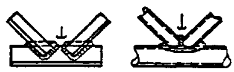
Case K_3 Severe stress concentration

Reference	Description	Figure	Symbol[1]
3,1	Parts of different thickness connected by butt welds (O.Q.) at right angles to the direction of the forces. 1 in 2 asymmetrical slope, or symmetrical position without blend slope		
3,11	Butt weld with backing strip and no backing run. Backing strip secured by intermittent tack welds		
3,12	Tubes joined by butt welds whose root is supported by a backing piece and not covered by a backing run		
3,13	Butt weld (O.Q.) at right angles to the direction of the forces at the intersection of flats with welded auxiliary gussets. The ends of the welds are ground, avoiding notches		

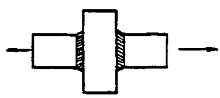
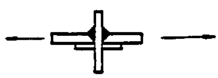
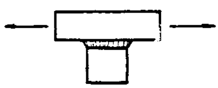
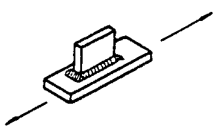
Case K_3 Severe stress concentration (continued)

Reference	Description	Figure	Symbol[(1)]
3,2	Continuous main member to which parts are fillet welded (O.Q.) at right angles to the direction of the forces. These parts take only a small portion of the loads transmitted by the main member		
3,21	Web and stiffener or transverse diaphragm secured by uninterrupted fillet weld (O.Q.)		
3,3	Continuous member to the edges of which are butt welded (O.Q.) parts parallel to the direction of the forces. These parts terminate in bevels and ends of the welds are ground avoiding notches		
3,31	Continuous member to which are welded parts parallel to the direction of the forces. These parts terminate in bevels or radii. Valid where the ends of the welds are fillet welds (S.Q.) over a length equal to 10 times the thickness, provided that the ends of the welds are ground, avoiding notches		
3,32	Continuous member through which extends a plate, terminating in bevels or radii parallel to the direction of the forces, secured by K-weld (O.Q.) over a length equal to 10 times the thickness		
3,33	Continuous member to which is welded a flat parallel to the direction of the forces, by means of fillet weld (S.Q.) in the indicated area when $e_1 < 1{,}5e_2$		
3,34	Members at the extremity of which connecting gussets are secured by a fillet weld (S.Q.) when $e_1 \leq e_2$. In case of unilateral gusset allow for eccentric load		
3,35	Continuous member to which stiffeners parallel to the direction of the forces are welded. The ends of the welds are fillet welds (S.Q.) over a length equal to ten times the thickness and are ground avoiding notches		

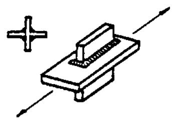
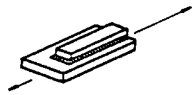
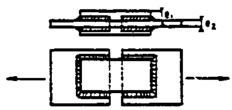
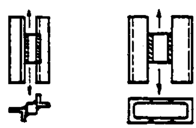
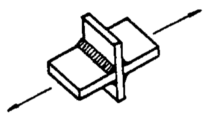
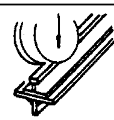
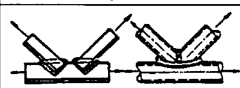
Case K_3 Severe stress concentration (continued)

Reference	Description	Figure	Symbol[1]
3,36	Continuous member to which stiffeners parallel to the direction of the forces are secured by fillet welds (O.Q.) which are intermittent or made between indentations		
3,4	Cruciform joint made with K-weld (O.Q.) at right angles to the direction of the forces		
3,41	K-weld (O.Q.) between flange and web in case of concentrated load in the plane of the web at right angles to the weld		
3,5	K-weld (O.Q.) joining parts stressed in bending and shear		
3,7	Continuous member to which sections or tubes are fillet welded (S.Q.)		

Case K_4 Very severe stress concentration

Reference	Description	Figure	Symbol[1]
4,1	Parts of different thickness butt welded (O.Q.) at right angles to the direction of the forces. Asymmetrical position without blend slope		
4,11	Butt welds (O.Q.) at right angles to the direction of the forces, at the intersection of flats (no auxiliary gussets)		
4,12	Single bevel weld at right angles to the direction of the forces, between intersecting parts (cruciform joint)		
4,3	Continuous member to the sides of which are welded parts ending at right angles, parallel to the direction of the forces		
4,31	Continuous member to which parts, ending at right angles, parallel to the direction of the forces, and receiving a large proportion of the loads transmitted by the main member, are secured by fillet weld (O.Q.)		

Case K_4 Very severe stress concentration (continued)

Reference	Description	Figure	Symbol[1]
4,32	Continuous member through which extends a plate ending at right angles and secured by fillet welding (O.Q.)		△
4,33	Continuous member on which a flat is secured by means of a fillet weld (O.Q.) parallel to the direction of the forces		◺
4,34	Joint plate secured by (O.Q.) fillet welds ($e_1 = e_2$). In case of unilateral joint plate allow for eccentric loads		◺
4,35	Parts welded one on the other secured by fillet welds (O.Q.) in a slot or in holes		
4,36	Continuous members between which connecting gussets are secured by fillet welds (O.Q.) or butt welds (O.Q.)		◺ >|
4,4	Cruciform joint made with fillet weld (O.Q.) at right angles to the direction of the forces		D △
4,41	Fillet weld (O.Q.) between flange and web in the case of concentrated load in the plane of the web at right angles to the weld		△
4,5	Fillet welds (O.Q.) joining parts stressed in bending and shear		D △
4,7	Continuous member to which sections or tubes are connected by fillet welds (O.Q.)		◺

The minimum and maximum stresses in the plates, profiles and connections have to be calculated.

$$\text{(kappa)} \quad \kappa = \frac{\sigma_{\min}}{\sigma_{\max}}$$

This leads to the allowed stress values, mentioned hereafter for groups E7 and E8.

Tension: κ is positive.
Compression: κ is negative.

(In kg/mm²)

TENSION AND COMPRESSION GROUP: E7 MATERIAL: Fe 360

	Non-welded members						Welded members									
	W0		W1		W2		K0		K1		K2		K3		K4	
Kappa	T	C	T	C	T	C	T	C	T	C	T	C	T	C	T	C
−1,0	13,32	13,32	11,32	11,32	9,32	9,32	10,35	10,35	9,24	9,24	7,76	7,76	5,54	5,54	3,33	3,33
−0,9	13,88	14,02	11,79	11,92	9,71	9,81	10,78	10,89	9,63	9,73	8,08	8,17	5,77	5,83	3,47	3,51
−0,8	14,48	14,80	12,30	12,58	10,13	10,36	11,25	11,50	10,04	10,27	8,43	8,62	6,02	6,16	3,62	3,70
−0,7	15,14	15,67	12,86	13,32	10,59	10,96	11,76	12,18	10,50	10,87	8,82	9,13	6,30	6,52	3,78	3,52
−0,6	15,86	16,65	13,48	14,15	11,10	11,65	12,32	12,94	11,00	11,55	9,24	9,70	6,60	6,93	3,96	4,16
−0,5	16,65	17,76	14,15	15,09	11,65	12,43	12,94	13,80	11,55	12,32	9,70	10,35	6,93	7,39	4,16	4,44
−0,4	17,53	19,03	14,89	16,17	12,26	13,31	13,62	14,79	12,16	13,20	10,21	11,09	7,29	7,91	4,38	4,75
−0,3	18,50	20,49	15,72	17,42	12,94	14,34	14,38	15,92	12,83	14.22	10,78	11,94	7,69	8,52	4,63	5,12
−0,2	19,59	22,20	16,65	18,87	13,71	15,53	15,22	17,25	13,59	15,40	11,41	12,93	8,15	9,23	4,90	5,55
−0,1	20,81	24,22	17,69	20,58	14,56	16,95	16,17	18,82	14,44	16,80	12,13	14,11	8,66	10,07	5,20	6,05
+0,0	22,20	26,64	18,87	22,64	15,53	18,64	17,25	20,70	15,40	18,48	12,93	15,52	9,23	11,08	5,55	6,66
+0,1	22,65	27,18	19,49	23,39	16,25	19,50	17,93	21,51	16,12	19,34	13,66	16,40	9,89	11,87	6,03	7,24
+0,2	23,13	27,75	20,16	24,19	17,03	20,44	18,66	22,39	16,90	20,29	14,48	17,38	10,66	12,79	6,61	7,93
+0,3	23,62	28,34	20,87	25,05	17,90	21,48	19,46	23,35	17,77	21,33	15,40	18,48	11,54	13,85	7,30	8,76
+0,4	24,13	28,96	21,64	25,96	18,85	22,62	20,33	24,39	18,74	22,48	16,45	19,73	12,60	15,11	8,16	9,79
+0,5	24,67	29,60	22,46	26,95	19,92	23,50	21,28	25,53	19,81	23,77	17,64	21,17	13,86	16,63	9,25	11,10
+0,6	25,23	30,27	23,35	28,02	21,11	25,33	22,32	26,78	21,01	25,21	19,03	22,84	15,40	18,48	10,67	12,81
+0,7	25,81	30,98	24,32	29,18	22,45	26,94	23,47	28,16	22,37	26,84	20,65	24,78	17,33	20,79	12,61	15,14
+0.8	26.43	31,71	25,36	30,43	23,98	28,77	24,74	29,69	23,91	28,70	22,58	27,09	19,81	23,77	15,42	18,50
+0,9	27,07	32,49	26,50	31,80	25,73	30,87	26,16	31,39	25,69	30,83	24,90	29,88	23,11	27,74	19,82	23,79
+1,0	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30

TENSION AND COMPRESSION GROUP: E7 MATERIAL: Fe 510

	Non-welded members						Welded members									
	W0		W1		W2		K0		K1		K2		K3		K4	
kappa	T	C	T	C	T	C	T	C	T	C	T	C	T	C	T	C
−1,0	13,66	13,66	11,62	11,62	9,57	9,57	10,35	10,35	9,24	9,24	7,76	7,76	5,54	5,54	3,33	3,33
−0,9	14,23	14,38	12,10	12,23	9,97	10,07	10,78	10,89	9,63	9,73	8,08	8,17	5,77	5,83	3,47	3,51
−0,8	14,85	15,18	12,63	12,91	10,40	10,63	11,25	11,50	10,04	10,27	8,43	8,62	6,02	6,16	3,62	3,70
−0,7	15,52	16,07	13,20	13,67	10,88	11,26	11,76	12,18	10,50	10,87	8,82	9,13	6,30	6,52	3,78	3,52
−0,6	16,26	17,08	13,83	14,53	11,39	11,96	12,32	12,94	11,00	11,55	9,24	9,70	6,60	6,93	3,96	4,16
−0,5	17,08	18,21	14,53	15,49	11,96	12,76	12,94	13,80	11,55	12,32	9,70	10,35	6,93	7,39	4,16	4,44
−0,4	17,97	19,51	15,29	16,60	12,59	13,67	13,62	14,79	12,16	13,20	10,21	11,09	7,29	7,91	4,38	4,76
−0,3	18,97	21,02	16,14	17,88	13,29	14,72	14,38	15,92	12,83	14,22	10,78	11,94	7,69	8,52	4,63	5,12
−0,2	20,09	22,77	17,09	19,37	14,07	15,95	15,22	17,25	13,59	15,40	11,41	12,93	8,15	9,23	4,90	5,55
−0,1	21,34	24,84	18,16	21,13	14,95	17,40	16,17	18,82	14,44	16,80	12,13	14,11	8,66	10,07	5,20	6,05
+0,0	22,77	27,32	19,37	23,24	15,95	19,14	17,25	20,70	15,40	18,48	12,93	15,52	9,23	11,08	5,55	6,66
+0,1	23,76	28,51	20,39	24,47	16,95	20,34	18,27	21,52	16,39	19,67	13,86	16,63	10,00	12,00	6,07	7,28
+0,2	24,83	29,80	21,53	25,84	18,09	21,71	19,42	23,30	17,52	21,02	14,93	17,91	10,90	13,08	6,70	8,04
+0,3	26,02	31,22	22,81	27,37	19,39	23,27	20,72	24,86	18,82	22,58	16,18	19,41	11,98	14,37	7,47	8,97
+0,4	27,31	32,78	24,25	29,10	20,89	25,07	22,20	26,64	20,32	24,38	17,65	21,18	13,29	15,95	8,45	10,14
+0,5	28,75	34,50	25,88	31,06	22,64	27,17	23,92	28,70	22,08	26,50	19,42	23,31	14,93	17,92	9,72	11,66
+0,6	30,35	36,41	27,75	33,30	24,71	29,66	25,92	31,11	24,18	29,01	21,59	25,91	17,03	20,44	11,43	13,72
+0,7	32,13	38,55	29,90	35,89	27,21	32,65	28,30	33,96	26,72	32,06	24,30	29,17	19,83	23,79	13,89	16,67
+0,8	34,13	40,96	32,43	38,91	30,26	36,31	31,15	37,37	29,85	35,82	27,80	33,35	23,71	28,45	17,68	21,22
+0,9	36,40	43,69	35,41	42,49	34,08	40,89	34,53	41,56	33,82	40,58	32,46	38,95	29,49	35,39	24,33	29,20
+1,0	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80

(In kg/mm²)

TENSION AND COMPRESSION GROUP: E8 MATERIAL: Fe 360

	Non-welded members						Welded members									
	W0		W1		W2		K0		K1		K2		K3		K4	
Kappa	T	C	T	C	T	C	T	C	T	C	T	C	T	C	T	C
−1,0	12,00	12,00	10,20	10,20	8,40	8,40	8,40	8,40	7,50	7,50	6,30	6,30	4,50	4,50	2,70	2,70
−0,9	12,50	12,63	10,63	10,74	8,75	8,84	8,75	8,84	7,81	7,89	6,56	6,63	4,69	4,74	2,81	2,84
−0,8	13,04	13,33	11,09	11,33	9,13	9,33	9,13	9,33	8,15	8,33	6,85	7,00	4,89	5,00	2,93	3,00
−0,7	13,64	14,12	11,59	12,00	9,55	9,88	9,55	9,88	8,52	8,82	7,16	7,41	5,11	5,29	3,07	3,18
−0,6	14,29	15,00	12,14	12,75	10,00	10,50	10,00	10,50	8,53	9,38	7,50	7,88	5,36	5,63	3,21	3,38
−0,5	15,00	16,00	12,75	13,60	10,50	11,20	10,50	11,20	9,38	10,00	7,88	8,40	5,63	6,00	3,38	3,60
−0,4	15,79	17,14	13,42	14,57	11,05	12,00	11,05	12,00	9,87	10,71	8,29	9,00	5,92	6,43	3,55	3,85
−0,3	16,67	18,46	14,17	15,69	11,67	12,92	11,67	12,92	10,42	11,54	8,75	9,69	6,25	6,92	3,75	4,15
−0,2	17,65	20,00	15,00	17,00	12,35	14,00	12,35	14,00	11,03	12,50	9,26	10,50	6,62	7,50	3,97	4,50
−0,1	18,75	21,82	15,94	18,55	13,13	15,27	13,13	15,27	11,72	13,64	9,84	11,45	7,03	8,18	4,22	4,91
+0,0	20,00	24,00	17,00	20,40	14,00	16,80	14,00	16,80	12,50	15,00	10,50	12,60	7,50	9,00	4,50	5,40
+0,1	20,57	24,69	17,69	21,22	14,73	17,68	14,73	17,68	13,23	15,87	11,20	13,44	8,09	9,71	4,91	5,89
+0,2	21,18	25,42	18,43	22,11	15,54	18,65	15,54	18,65	14,04	16,85	11,99	14,39	8,78	10,54	5,41	6,49
+0,3	21,83	26,19	19,24	23,08	16,44	19,73	16,44	19,73	14,97	17,95	12,91	15,49	9,60	11,52	6,01	7,21
+0,4	22,52	27,02	20,12	24,14	17,46	20,95	17,46	20,95	16,02	19,23	13,97	16,77	10,59	12,71	6,77	8,12
+0,5	23,25	27,90	21,08	25,30	18,61	22,33	18,61	22,33	17,24	20,68	15,24	18,28	11,81	14,17	7,74	9,29
+0,6	24,03	28,83	22,15	26,58	19,92	23,91	19,92	23,91	18,65	22,36	16,75	20,09	13,34	16,01	9,05	10,86
+0,7	24,86	29,83	23,33	27,99	21,43	25,72	21,43	25,72	20,31	24,38	18,59	22,31	15,33	18,40	10,88	13,06
+0,8	25,75	30,50	24,63	29,56	23,19	27,83	23,19	27,83	22,31	26,77	20,89	25,06	18,02	21,62	13,65	16,38
+0,9	26,71	32,06	26,10	21,32	25,27	30,32	25,27	30,32	24,73	29,68	23,83	28,60	21,85	26,22	18,30	21,96
+1,0	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30	27,75	33,30

TENSION AND COMPRESSION GROUP: E8 MATERIAL: Fe 510

	Non-welded members						Welded members									
	W0		W1		W2		K0		K1		K2		K3		K4	
Kappa	T	C	T	C	T	C	T	C	T	C	T	C	T	C	T	C
−1,0	12,00	12,00	10,20	10,20	8,40	8,40	8,40	8,40	7,50	7,50	6,30	6,30	4,50	4,50	2,70	2,70
−0,9	12,50	12,63	10,63	10,74	8,75	8,84	8,75	8,84	7,81	7,89	6,56	6,63	4,69	4,74	2,81	2,84
−0,8	13,04	13,33	11,09	11,33	9,13	9,33	9,13	9,33	8,15	8,33	6,85	7,00	4,89	5,00	2,93	3,00
−0,7	13,64	14,12	11,59	12,00	9,55	9,88	9,55	9,88	8,52	8,82	7,16	7,41	5,11	5,29	3,07	3,18
−0,6	14,29	15,00	12,14	12,75	10,00	10,50	10,00	10,50	8,93	9,38	7,50	7,88	5,36	5,63	3,21	3,38
−0,5	15,00	16,00	12,75	13,60	10,50	11,20	10,50	11,20	9,38	10,00	7,88	8,40	5,63	6,00	3,38	3,60
−0,4	15,79	17,14	13,42	14,57	11,05	12,00	11,05	12,00	9,87	10,71	8,29	9,00	5,92	6,43	3,55	3,86
−0,3	16.67	18,46	14,17	15,69	11,67	12,92	11,67	12,92	10,42	11,54	8,75	9,69	6,25	6,92	3,75	4,15
−0,2	17,65	20,00	15,00	17,00	12,35	14,00	12,35	14,00	11,03	12,50	9,26	10,50	6,62	7,50	3,97	4,50
−0,1	18,75	21,82	15,94	18,55	13,13	15,27	13,13	15,27	11,72	13,64	9,84	11,45	7,03	8,18	4,22	4,91
+0,0	20,00	24,00	17,00	20,40	14,00	16,80	14,00	16,80	12,50	15,00	10,50	12,60	7,50	9,00	4,50	5,40
+0,1	21,02	25,23	18,02	21,62	14,96	17,55	14,96	17,95	13,41	16,09	11,33	13,59	8,16	9,79	4,94	5,92
+0,2	22,16	26,59	19,16	22,99	16,06	19,27	16,06	19,27	14,47	17,36	12,30	14,76	8,94	10,73	5,47	6,56
+0,3	23,42	28,11	20,46	24,56	17,33	20,80	17,33	20,80	15,70	18,84	13,45	16,14	9,90	11,88	6,13	7,35
+0,4	24,84	29,81	21,95	26,34	18,83	22,59	18,83	22,59	17,17	20,60	14,84	17,80	11,08	13,30	6,96	8,36
+0,5	26,44	31,73	23,68	28,41	20,60	24,72	20,60	24,72	18,93	22,72	16,55	19,85	12,58	15,10	8,07	9,68
+0,6	28,26	33,91	25,70	30,84	22,75	27,30	22,75	27,30	21,10	25,32	18,70	22,44	14,55	17,46	9,59	11,51
+0,7	30,35	36,42	28,09	33,71	25,40	30,47	25,40	30,47	23,84	28,61	21,50	25,80	17,26	20,71	11,82	14,18
+0,8	32,77	39,33	30,98	37,18	28,74	34,48	28,74	34,48	27,39	32,87	25,28	30,33	21,20	25,43	15,39	18,47
+0,9	35,62	42,74	34,53	41,44	33,09	39,71	33,09	39,71	32,18	38,61	30,67	36,81	27,46	32,96	22,08	26,49
+1,0	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80	39,00	46,80

For forestays, hangers and backstays, it is advised that the tension allowed should be decreased to 0,6 of the usual allowance for fatigue tension in such parts.

The Rainflow method

BS 5400, Part 10, 1980 gives another method for calculating fatigue in steel structures. An excerpt of this method is reproduced with the permission of the British Standards Institution under license number 2001/SK0362. Complete standards can be obtained from BSI Customer Services (Tel. 0044 20 8996 9001).

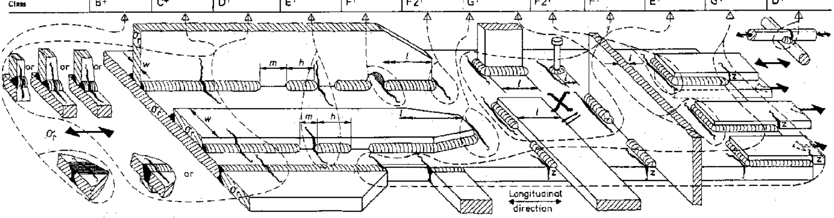
Charts for classification of details for fatigue BS 5400 : Part 10C : 1980

(extracted from BS 5400 Steel, concrete and composite bridges Part 10. Code of practice for fatigue)

Table 17. Classification of details

(b) Welded details on surface of member

Product form	Rolled steel structural plates, sections and built-up members											Reinforcing steel in concrete
Location of potential crack initiation	At a long welded attachment (in direction of σ_r)								At a short welded attachment		At any attachment	At welded intersections in fabric or between hot rolled bars
	Away from weld end			At an intermediate gap in a longitudinal weld	At a cope hole	At a weld end						
					Narrow attachment		Wide attachment				Close to edge of member	
							On one side only	On both sides symmetrically				
Dimensional requirements	Butt weld full penetration			Fillet weld		Weld toe not less than 10 mm from member edge					Weld toe within 10 mm of member edge	
				Intermittent $\frac{m}{h} < 2.5$		Weld length (parallel to σ_r) $l > 150$ mm			$l < 150$ mm			
					Attachment width $w \leq 50$ mm		$w > 50$ mm					
Manufacturing requirements (see also Part 6)	Grind smooth any undercut on member edges										Grind any undercut	Resistance or manual plus grind smooth undercut
	Dress flush reinforcement	Automatic no stop-starts									Avoid weld returns round laps (see z)	
Special inspection requirements	Proved free of all significant defects											
Design stress area	Minimum transverse cross section of member at location of potential crack initiation											
Special design stress parameter									$\tau \leq 0.5\,\sigma$	$\tau > 0.5\,\sigma$		
Type number	2.1*	2.2*	2.3	2.4*	2.5	2.6	2.7*	2.8*	2.9	2.10*	2.11*	2.12
Class	B‡	C‡	D†	E†	F†	F2†	G†	F2†	F†	E‡	G†	D†



Key. Typical crack location Surface grinding Direction of stress fluctuation

NOTE. Weld throat cracks are type 3 (see table 17(c)). *See H.3.3. †Important features that change significantly from one type to another. ‡Classifications that should be used with caution.

British Standards Institution · 2 Park Street London W1A 2BS Telephone 01-629 9000 · Telex 266933 [Set of three charts. Chart (b)] Gr 5 + VAT

APPENDIX A

Basis of $\sigma_t - N$ relationship

A.1 General. The $\sigma_r - N$ relationships have been established from statistical analyses of available experimental data (using linear regression analysis of log σ_t and log N) with minor empirical adjustments to ensure compatability of results between the various classes.

The equation given in 11.2 may be written in basic form as:

$$N \times \sigma_r^m = K_0 \times \Delta^d$$

where

N is the predicted number of cycles to failure of a stress range σ_r

K_0 is the constant term relating to the mean-line of the statistical analysis results

m is the inverse slope of the mean-line $\log \sigma_r - \log N$ curve

Δ is the reciprocal of the anti-log of the standard deviation of $\log N$

d is the number of standard deviations below the mean-line.

NOTE. This corresponds to a certain probability of failure as shown in Table 10.

The relevant values of these terms are given in Tables 9 and 10 and the mean-line relationships are plotted in Fig. 15.

A.2 Treatment of low stress cycles. Under fluctuating stress of constant amplitude, there is a certain stress range below which an indefinitely large number of cycles can be sustained. The value of this 'non-propagating stress range' varies both with the environmental and with the size of any initial defect in the stressed material. In clean air, a steel detail which complies with the requirements of Parts 6, 7 or 8 is considered to have a constant amplitude non-propagating range σ_0 equal to the value of σ_r obtained from the formula in A.1 when $N = 10^7$.

Table 9 Mean-line $\sigma_r - N$ relationships

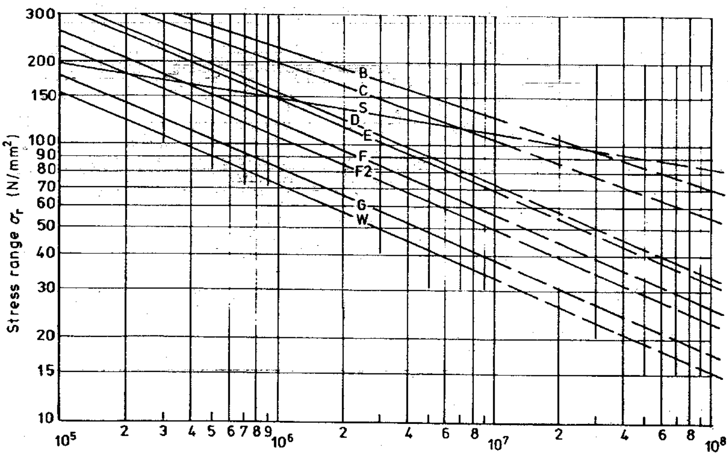
Detail class	K_0	Δ	m
W	$0{,}37 \times 10^{12}$	0,654	3,0
G	$0{,}57 \times 10^{12}$	0,662	3,0
F2	$1{,}23 \times 10^{12}$	0,592	3,0
F	$1{,}73 \times 10^{12}$	0,605	3,0
E	$3{,}29 \times 10^{12}$	0,561	3,0
D	$3{,}99 \times 10^{12}$	0,617	3,0
C	$1{,}08 \times 10^{14}$	0,625	3,5
B	$2{,}34 \times 10^{15}$	0,657	4,0
S	$2{,}13 \times 10^{23}$	0,313	8,0

Table 10 Probability factors

Probability of failure	d
50%	0*
31%	0,5
16%	1,0
2,3%	2,0†
0,14%	3,0

* Mean-line curve.

† The standard design curve of 11.2.



NOTE. The use of these curves for calculation purposes is not recommended.

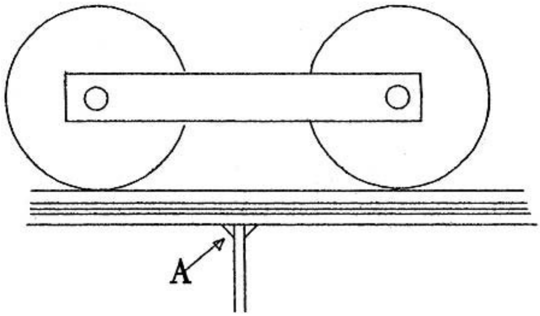
Fig. 15. Summary of mean-line σ_r – N curves

When the applied fluctuating stress has varying amplitude, so that some of the stress ranges are greater and some less than σ_0, the larger stress ranges will cause enlargement of the initial defect. This gradual enlargement reduces the value of the non-propagating stress range below σ_0. Thus, as time goes on, an increasing number of stress ranges below σ_0 can themselves contribute to the further enlargement of the defect. The final result is an earlier fatigue failure than could be predicted by assuming that all stress ranges below σ_0 are ineffective.

This phenomenon has been studied on principles derived from fracture mechanics. It is found that an adequate approximation to the fatigue performance so predicted can be obtained by assuming that a certain fraction $(\sigma_r/\sigma_0)^2$ of stress ranges σ_r less than σ_0 cause damage in accordance with the formula in A.1.

Example

1. Calculate the maximum and minimum stress in point A.
   Assume the maximum stress is $\sigma_{max} = 18\,kN/cm^2$ and the minimum stress is $\sigma_{min} = 8\,kN/cm^2$.
   The stress range is then: $\sigma_r = (18 - 8)\,kN/cm^2$ or $100\,N/mm^2$.
2. Check how many stress variations can be expected during the lifetime of the crane. Assume that this is 2 000 000 cycles, or the endurance according to Fig. 15 of BS 5400 is then $N = 2.10^6$ cycles.
3. Check in Table 17 of BS 5400 which part of the figure has to be taken into account. We choose Part X.



4. Now it can be read from Fig. 15 under Endurance $N = 2.10^6$ cycles and line F2 that the allowed stress range $\sigma_r = 85\,N/mm^2$, so less than the calculated $\sigma_r = 100\,N/mm^2$ (we do not take line F, as we weld continuously).

   This indicates that we are in a dangerous area. A redesign has to be done and measures have to be taken, in order to get the stress range σ_r under the allowed $\sigma_r = 85\,N/mm^2$.

Tubes

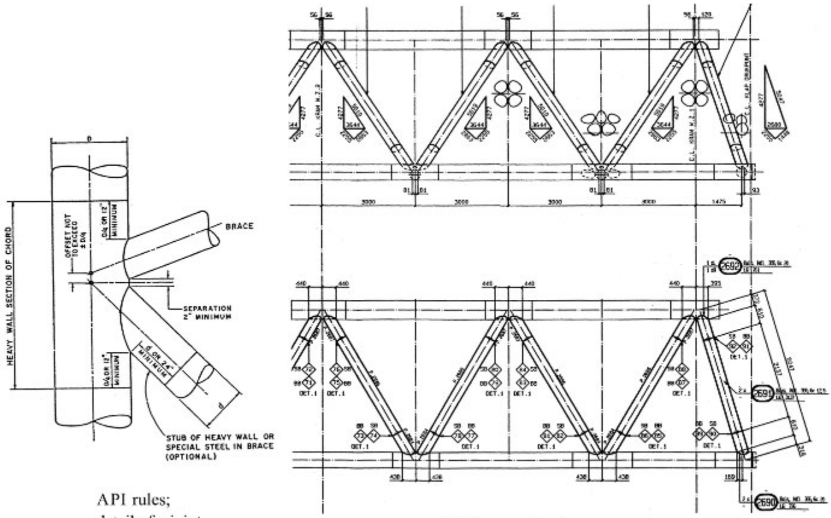
Tubes are normally influenced by loads, giving tension or compression (and eventually bending) stresses in the tubes. These stresses in the tubes can be calculated with the formulae mentioned in standards like DIN and FEM, or with the formulae of the API rules, etc.

However, the brace thickness and the chord thickness in tubular joints without gussets should be best calculated in accordance with the methods, mentioned in the API rules (American Petroleum Institute; 1220 L. Street, Northwest, Washington, DC 20005), as the influence of fatigue can be such that the brace thickness and the chord thickness must be locally increased in order to avoid damage (Fig. 7.5.9).

C – Fatigue in mechanism components as shafts etc.

The following, simple way of calculation is recommended:

– calculate the bending and torsion stresses out of the nominal kWs from the motor(s) and/or from the nominal wheel loads and torques in the wheel shafts.



API rules;
detail of a joint

Lattice construction

Fig. 7.5.9

– check the allowable stresses in N/mm^2 in Table 7.6.2;
– check kappa (κ).

$$\kappa = \frac{\sigma_{\min}}{\sigma_{\max}} \quad \text{or} \quad \kappa = \frac{\tau_{\min}}{\tau_{\max}} \text{ in the case of shear and torsion}$$

This ratio can vary from –1 to +1 and is positive if the extreme stresses are both of the same type, fluctuating stresses, and negative when the extreme stresses are of opposite types, alternating stresses.

Examples

1. **Conical shaft with key**

   Heavy duty

   Torsion: Fluctuating, $\kappa = +1$

   Shaft material 34 Cr Ni Mo6

   $\tau = 135$ N/mm^2

   $\beta_{KW} = 1{,}5$

   $$\tau_{all} = \frac{135}{1{,}5} = 90 \text{ N/mm}^2$$

Table 7.6.2 Allowable stresses in N/mm²

Material	Way of loading / Stresses	Normal duty ($s = 1{,}5$) Compression; tension bending	Normal duty ($s = 1{,}5$) Torsion	Heavy duty ($s = 1{,}8$) Compression; tension bending	Heavy duty ($s = 1{,}8$) Torsion	Very heavy duty ($s = 2{,}16$) Compression; tension bending	Very heavy duty ($s = 2{,}16$) Torsion
Fe 360	Fluctuating $\kappa = +1$	135	63	112	52	94	44
	Alternating $\kappa = -1$	77	45	64	37	54	31
Fe 510 CK 45	Fluctuating $\kappa = +1$	186	80	155	66	129	55
	Alternating $\kappa = -1$	108	59	90	49	75	41
CK 45	Fluctuating $\kappa = +1$	204	87	170	73	141	60
	Alternating $\kappa = -1$	113	63	94	53	79	44
CK 60	Fluctuating $\kappa = +1$	227	98	190	82	158	68
	Alternating $\kappa = -1$	124	72	104	60	86	50
42 Cr Mo4	Fluctuating $\kappa = +1$	335	134	279	112	233	93
	Alternating $\kappa = -1$	215	97	179	81	150	67
34 Cr Ni–Mo6	Fluctuating $\kappa = +1$	385	162	321	135	267	112
	Alternating $\kappa = k = -1$	236	111	197	93	164	77

2. **Shaft with reduced diameters**

Normal duty

Bending: Alternating, $\kappa = -1$

Shaft material 42 Cr Mo4

$\sigma = 215\ \mathrm{N/mm^2}$

$$D/d = \frac{70}{50} = 1{,}4 \qquad C = 0{,}64$$

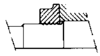
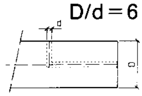
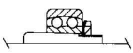
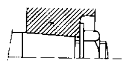
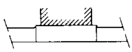
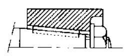
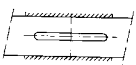
$$r/d = \frac{10}{50} = 0{,}2 \qquad \beta_{\mathrm{KO}} = 1{,}30$$

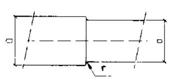
$$\beta_{\mathrm{K}} = 1 + C \cdot (\beta_{\mathrm{KO}} - 1) = 1 + 0{,}64(1{,}30 - 1)$$

$$\beta_{\mathrm{K}} = 1{,}19$$

$$\sigma_{\mathrm{all}} = \frac{215}{1{,}19} = 180\ \mathrm{N/mm^2}$$

Table 7.6.3 Factors in mechanical components

Construction:	Tolerance	β_{kb}	β_{kw}		Tolerance	β_{kb}	β_{kw}
 Shaft with thread		1,5	1,1	D/d = 6  Shaft with grease hole		1,7	1,4
 Bearing on shaft	H7/k6	1,6	1,25	 Con. shaft without key		1,8	1,35
 Wheel, pressed on shaft	H7/p6 H7/s6	1,8	1,35	 Con. shaft with key		2,2	1,5
 Shaft with 1 key	H7/m6	2,0	1,4	 Shaft: press-keyed		1,5	1,15



$r/d = 0.033$ $\beta_{ko} = 2,15$
$r/d = 0,10$ $\beta_{ko} = 1,50$
$r/d = 0,20$ $\beta_{ko} = 1,30$

D/d	1,0	1,1	1,2	1,3	1,4	1,5	1,6	1,7	1,8	1,9	2,0
C	0	0,20	0,41	0,53	0,64	0,73	0,82	0,87	0,92	0,97	1,0

$\beta_k = 1 + C\ (\beta_{ko^{-1}})$ shaft with reduced diameters

Check of the safety coefficient v in cases that bending and torsion is acting on a shaft: shaft with 1 key

Normal duty ($S = 1{,}5$)

Shaft material 42 Cr Mo4

Bending: Alternating $\kappa = -1$ $\quad \sigma_{\text{in shaft}} = 60\ \text{N/mm}^2$

Torsion: Fluctuating $\kappa = +1$ $\quad \tau_{\text{in shaft}} = 40\ \text{N/mm}^2$

Table: $\beta_{kb} = 2{,}0$ $\beta_{KW} = 1{,}4$

Table: $\sigma = 215\ \text{N/mm}^2$ $\tau = 134\ \text{N/mm}^2$

$$\sigma_{all} = \frac{215}{2} = 107{,}5\ \text{N/mm}^2$$

$$\tau_{all} = \frac{134}{1{,}4} = 95{,}7\ \text{N/mm}^2$$

$$v^1 = \frac{\sigma_{all}}{\sigma_{\text{in shaft}}} = \frac{107{,}5}{60} = 1{,}79$$

$$v_{1\sigma} = v^1 \cdot s = 1{,}79 \cdot 1{,}5 = 2{,}68$$

$$v^{11} \frac{\tau_{all}}{\tau_{\text{in shaft}}} = \frac{95{,}7}{40} = 2{,}39$$

$$v_{2\tau} = v^{11} \cdot s = 2{,}39 \cdot 1{,}5 = 3{,}58$$

$$v_{res} = \frac{1}{\sqrt{\frac{1}{(v_{1\sigma})^2} + \frac{1}{(v_{2\tau})^2}}} = \frac{1}{\sqrt{\frac{1}{2{,}68^2} + \frac{1}{3{,}58^2}}} = \frac{1}{0{,}46} = 2{,}17$$

v_{res} must be bigger than the safety coefficient.
$s = 1{,}5$ (Here for normal duty). This is the case.

Pressure between shafts and steel constructions

The maximum allowed pressures between shafts and the steel constructions in hinge points, travelling mechanisms, and so on, can be taken as follows, for:

- fluctuating loads: combined from vertical and horizontal loads;
- non-rotating shafts;
- shaft material Fe 510 (S355);
- steel construction material Fe 510 (S355)

F = max. load in Newtons
d = shaft diameter in mm
B = actual load carrying thickness in mm

$$\sigma_v = \frac{F \cdot l}{d \cdot B}\ \text{N/mm}^2$$

Allowed pressures

- Normal duty $\sigma_v = 100\ \mathrm{N/mm^2}$
- Heavy duty $\sigma_v = 85\ \mathrm{N/mm^2}$
- Very heavy duty $\sigma_v = 70\ \mathrm{N/mm^2}$

For alternating loads these figures have to be diminished by 30 percent.

D – Design details

In constructions which are being exposed to fatigue loads a number of design details can often be avoided to prevent fatigue cracks.

The following figures give some of these details.

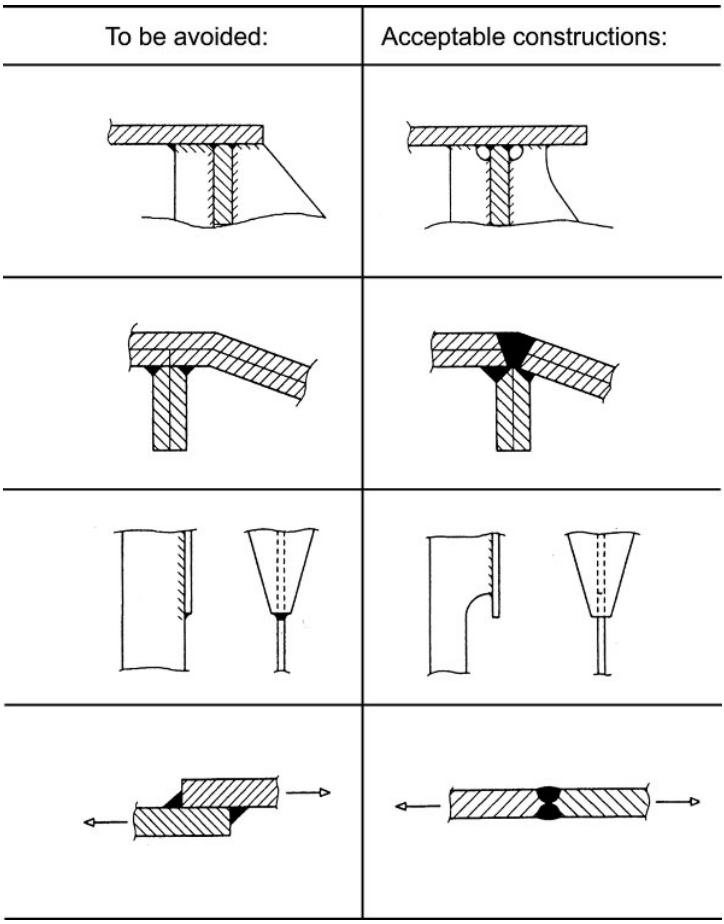


Fig. 7.6.1 Details of fatigue-sensitive constructions

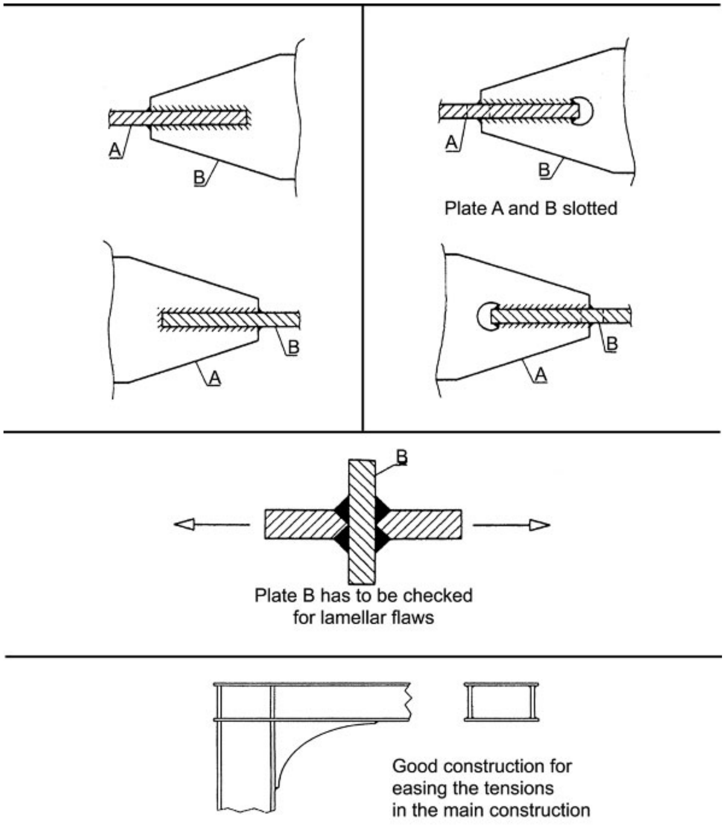


Fig. 7.6.1 Continued

7.7 The natural frequency

The natural frequency of a crane is a very important subject. It is necessary to calculate this figure in order to know whether a crane is stiff enough, flexible or even shaky.

For cranes with fast running trolleys the natural frequency, in trolley travel direction, has to be controlled.

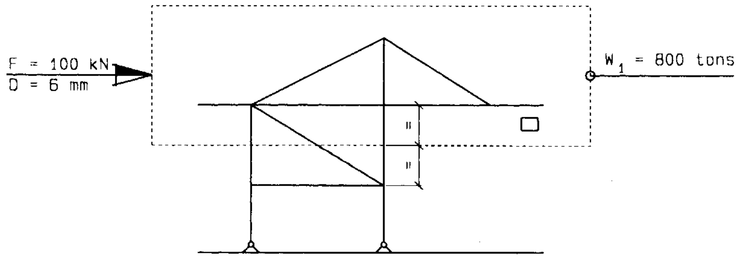


Fig. 7.7.1 The natural frequency

How to control this

- Using a computer, make a calculation of the displacement in the horizontal direction of the main- or trolley-girders of the crane under a particular horizontal force.

Assuming the following:

- weight of complete upperstructure, plus half of the underportal, plus trolley and load: $W = 800$ tons
- calculated displacement in horizontal direction under load: $D = 6\,\text{mm} = 0{,}006\,\text{m}$ (measured from quay level up to centre of main girders).

Then $C = \dfrac{F}{D} = \dfrac{100}{0{,}006} = 16\,666\,\text{kN/m}$

Frequency $\omega = \sqrt{\dfrac{C}{W_1}} = \sqrt{\dfrac{16\,666}{800}} = 4{,}56\,\text{rad/sec}$

Natural frequency $f = \dfrac{\omega}{2\pi} = \dfrac{4{,}56}{2\pi} = 0{,}726\,\text{Hz}$

Vibration time $t = \dfrac{1}{f} = \dfrac{1}{0{,}726} = 1{,}38\,\text{sec}$

In order to have a good reasonably stiff crane, the natural frequency should be in the range of $f = 0{,}70$ Hz.

Chapter 8

Wheels and Tracks

8.1 Calculating the wheel diameters of fast-running trolleys ($v > 100$ m/min)

The best way to calculae the diameter of the trolley travelling wheel is as follows:

- Calculate the maximum wheel load R maximum (tons).
- Choose a rail width and the material for the rail, being:

Fe510 – (St50):	for a rail, welded to the construction.
Fe600 – (St60) or	
Fe710 – (St70):	for a special, forged rail material.
for Fe510:	$P_{all} = 50$ kg/cm^2
for Fe600/Fe710:	$P_{all} = 60$ kg/cm^2
rail width:	K (cm)
radius of the curvature of the rail sides:	r (cm)

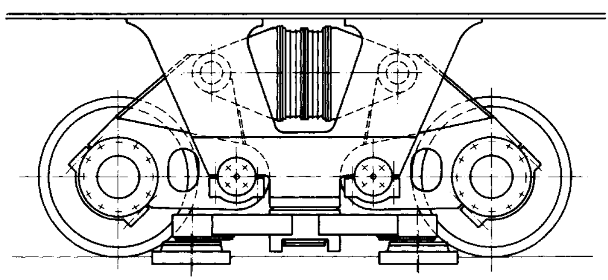


Fig. 8.1.1 Heavy-duty trolley bogie

D_{wheel} wheel diameter (cm)

$$D_{\text{wheel}} = \frac{R \cdot 1000}{P_{\text{all}} \cdot (K - 2 \cdot r)} \text{ cm}$$

For the hardness of the rims of the wheels, see under Section 8.2.

8.2 Calculating the wheel diameter of a crane travelling wheel for normal speeds (v = up to 60 m/min)

Calculate the average wheel load as follows:

$$R_{\text{mean}} = \frac{2 \cdot R_{\max} + R_{\min}}{3}$$

where

R_{mean} = average wheel load (tons)

$R_{\max}$ = maximum wheel load (tons)

$R_{\min}$ = minimum wheel load (tons)

C = rating factor, considered over one hour of crane working time.

Herein is: rating	40 percent:	$C = 1$
	60 percent	$C = 0{,}9$
	80 percent	$C = 0{,}8$

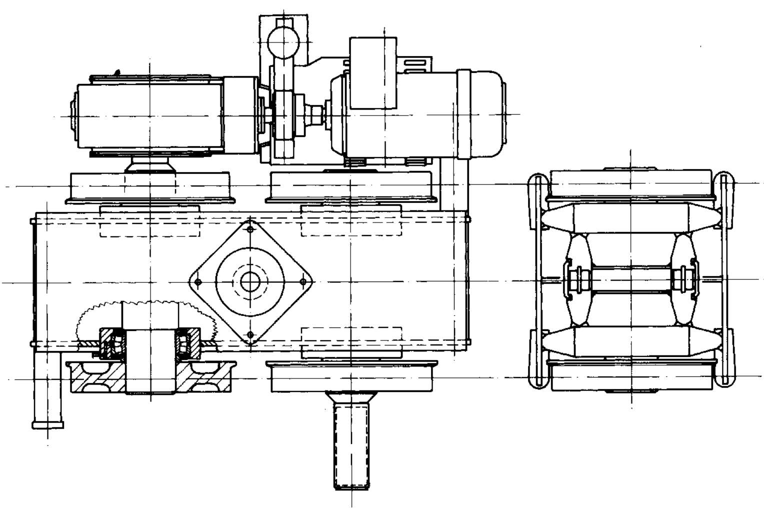


Fig. 8.2.1 Crane travelling bogie with double rails

P_{all}:	70 kg/cm^2 for a forged rail of Fe600 or Fe700
Rail width:	K (cm)
Radius of the curvature of the rail:	r (cm)
D_{wheel}:	wheel diameter

$$D_{wheel} = \frac{R_{mean} \cdot 1000}{C \cdot P_{all} \cdot (K - 2 \cdot r)}$$

Also check what the maximum static wheel load under the worst condition is, and divide this by 1,25. If the value is bigger than R_{mean}, then augment D_{wheel} accordingly.

The hardness of the rims of the crane travelling wheels and trolley travelling wheels should be approximately 300 HBr.

Delachaux in Gennevilliers, France, has 'infatigable' wheels with a rim which is deep hardened to 400/450 HBr. The depth of hardness can be as much as 20 mm.

8.3 Differences in wheel load, due to braking forces

Assume that the *horizontal* windload *per corner* is X tons. The X tons give this horizontal force to the main hinge point of the bogie train.

Fig. 8.3.1 Crane travelling mechanism

(X • 950) : 4800 = 0.2 X
(1/3X • 660) : 2400 = 0.09X
(1/3X • 876) : 1200 = 0.24X
(2/3X • 660) : 2400 = 0.18X

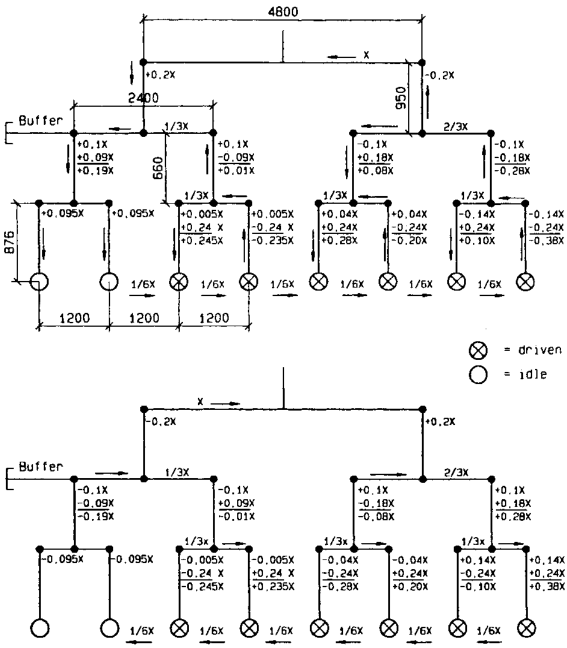


Fig. 8.3.2 Wheel load scheme through the horizontal force *X*

This horizontal force X results in vertical forces on the wheels. Assuming further, that eight wheels are under each corner, six of which are driven. As six wheels are driven and braked, these driven wheels can take $X:6 = \frac{1}{6}X$ as horizontal force. The scheme in Fig. 8.3.2 shows this phenomenon. The figures show which vertical wheel load should be added or subtracted per wheel, as well as the horizontal force per braked wheel.

Note: In Sections 8.1 and 8.2 the materials have been mentioned in the well-used nomenclature. The new nomenclature has been mentioned in Section 7.1.

Fe510 (St50) is S355
Fe600 (St60) is S335
Fe710 (St70) is S360

8.4 Rails and rail constructions

As mentioned earlier the block-rail of the material Fe510 (S355) is very popular for trolley travelling rails when the rails are welded to the girders. Rails with a higher strength are often more difficult to weld.

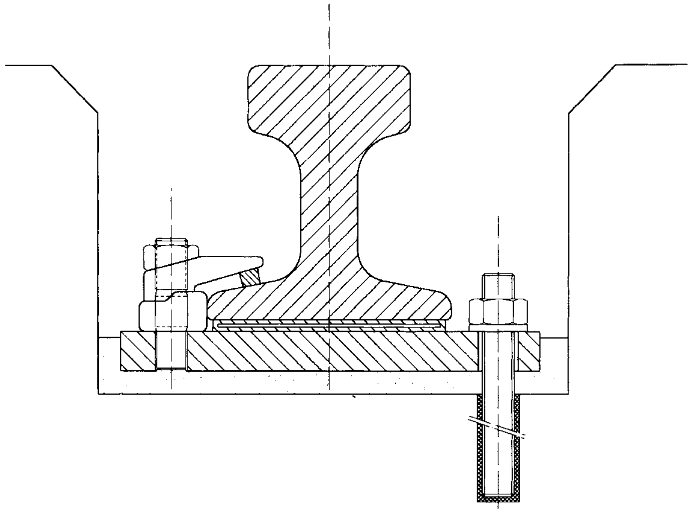
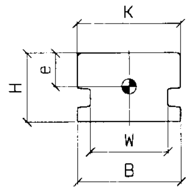


Fig. 8.4.1 Typical crane rail construction

Fig. 8.4.2 Crane travelling rails on sleepers

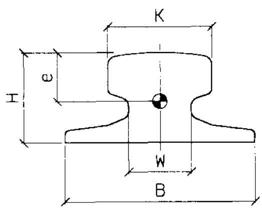
Table 8.4.1 Dimensions, etc. of rails



TYPE F

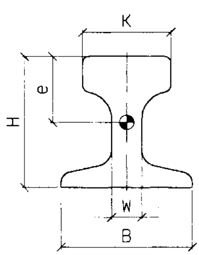
Rail type	Weight G (kg/m)	Height H (mm)	Base B(mm)	Head width K (mm)	Web W (mm)	Mom. of inertia J_x (cm^4)	Section modulus W_x (cm^3)	Neutral axis e (mm)
F100	57,5	80	100	100	70	414	101	39,1
F120	70,1	80	120	120	90	499	123	39,3

Material: steel with a tensile strength of min. 690 N/mm^2



TYPE A

A55	32	65	150	55	31	182	46,9	38,8
A65	43,5	75	175	65	38	327	73,6	44,4
A75	56,5	85	200	75	45	545	109	50
A100	75,2	95	200	100	60	888	170	52,1
A120	101	105	220	120	72	1420	249	57
A150	150	150	220	150	80	4373	412	72



BSC164	166	150	230	140	75	4777		
MRS125	125	180	180	120	40	6225	681	91,4
MR151	151	150	220	150	80	4271	560	73,7
MRS220	221	160	220	220	115	6890		
PR185 R	86,8	152,4	152,4	101,6	34,9	3068	399	76,8

Table 8.4.2 Rail qualities

Steel grade (N/mm^2)	*C*	*Mn*	*Si*	S_{max}	P_{max}	*Va*	σ_e (N/mm^2)
690	0,410–0,520	0,90–1,10	0,15–0,25	0,050	0,060	—	360
780	0,490–0,620	1,10–1,30	0,20–0,30	0,050	0,050	—	400
880	0,580–0,720	1,30–1,60	0,25–0,40	0,050	0,050	—	450
690 Va	0,260–0,380	1,10–1,30	0,20–0,30	0,050	0,050	0,10–0,15	460
780 Va	0,320–0,420	1,10–1,30	0,25–0,35	0,050	0,050	0,10–0,15	500
880 Va	0,420–0,500	1,20–1,40	0,25–0,30	0,050	0,050	0,10–0,15	560

Fig. 8.4.3 Barge cranes

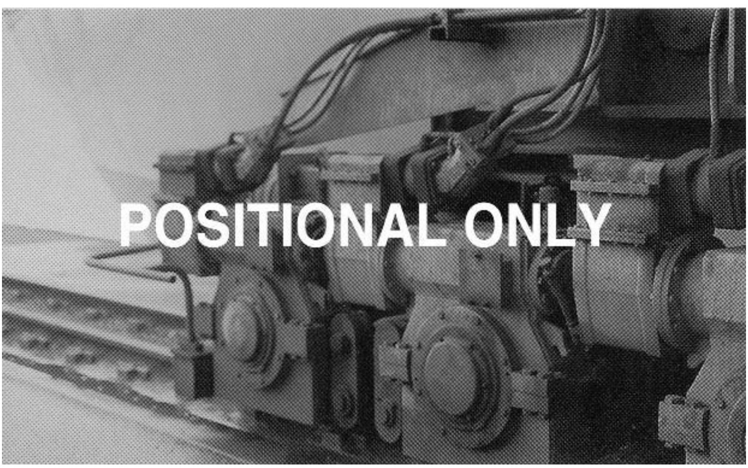


Fig. 8.4.4 Hydraulic-driven crane travelling mechanism in an ore unloader

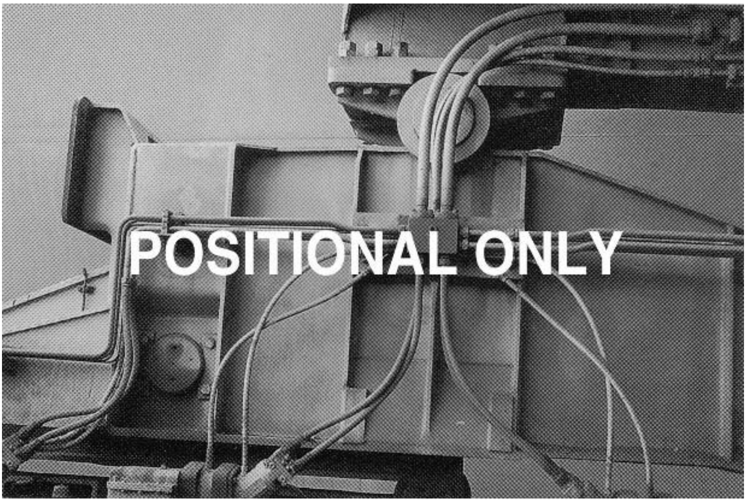


Fig. 8.4.5 Feeding the crane travelling motors of Fig. 8.4.4

Forged crane rails exist in many types, only some of which are mentioned here. The newer crane rails often have a crowned rail head (R = 600 mm).

8.5 Trolley travelling rails and boom hinge points

Trolley travelling rails

Trolley travelling rails are often welded to the girders and the booms. In this case a block-type rail should be used, preferably of steel quality Fe510 (S355) which has a low carbon content.

Heavy duty constructions, permitting high wheel-loads can be made with forged crane rails. These rails include C, Mn, Si. Vanadium could be included when the loads and the frequency of overrollings are high.

Fastening these rails to the girders should be done with clips. The rails are then normally laid on a flexible pad.

Boom hinge points

The boom hinge points are rather vulnerable. Many different types are used. It is most important that the construction is stiff, and that the load carrying parts should not be able to deflect. If the rail components in the hinge points are hardened, this is a useful feature.

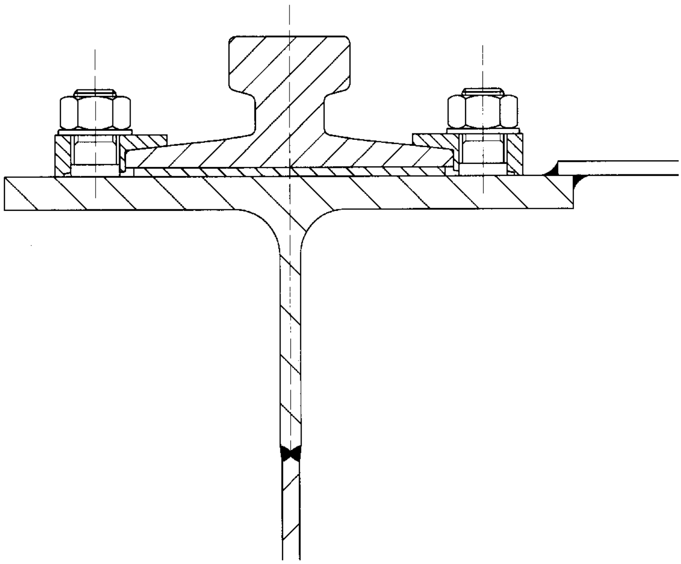


Fig. 8.5.1 Clipped trolley rail

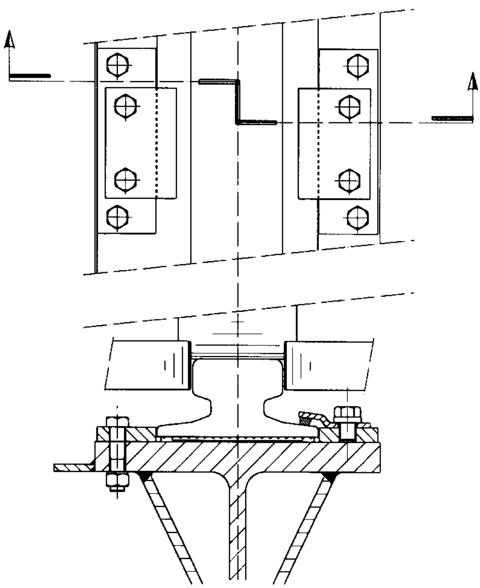


Fig. 8.5.2 Very heavy-duty construction

Fig. 8.5.3 Boom hinge point in heavy-duty ore unloader

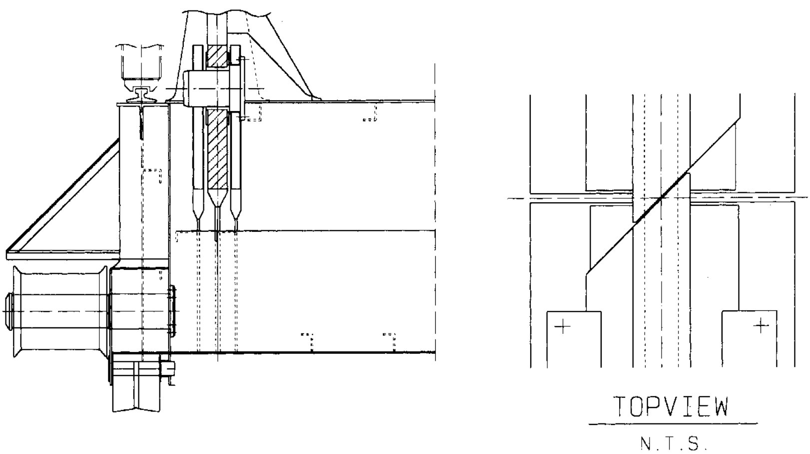


Fig. 8.5.4 Boom hinge point of a container crane

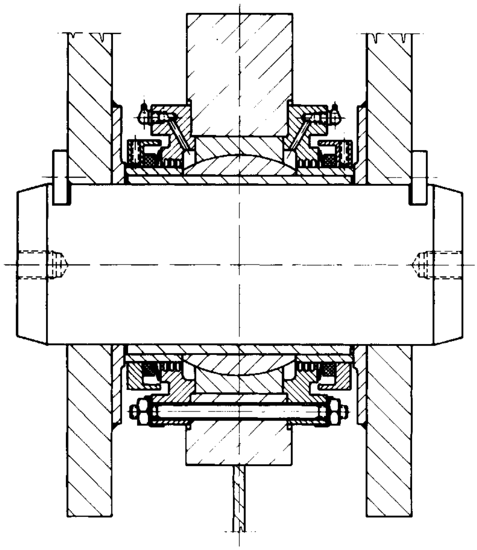


Fig. 8.5.5 Special construction of a hinge point

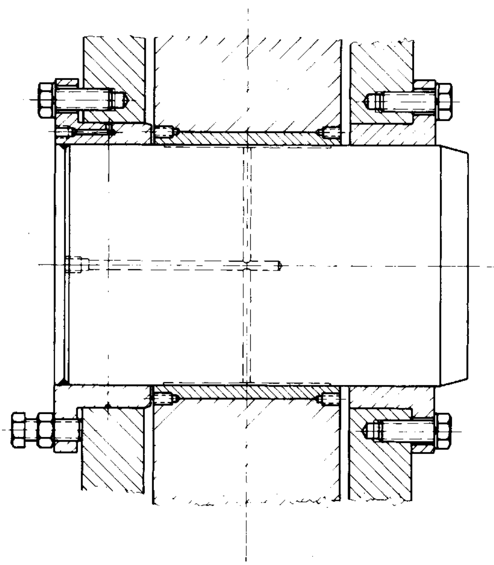


Fig. 8.5.6 Bronze bushed hinge point

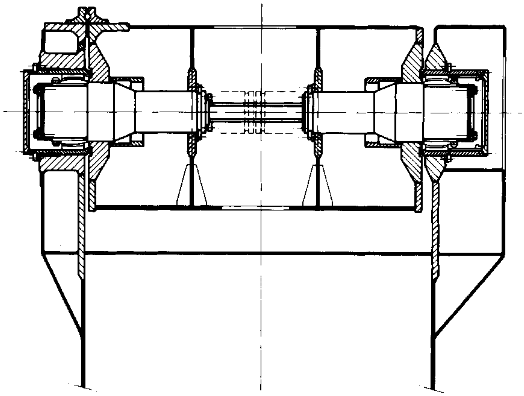


Fig. 8.5.7 Detail of hinge point of Fig. 8.5.3

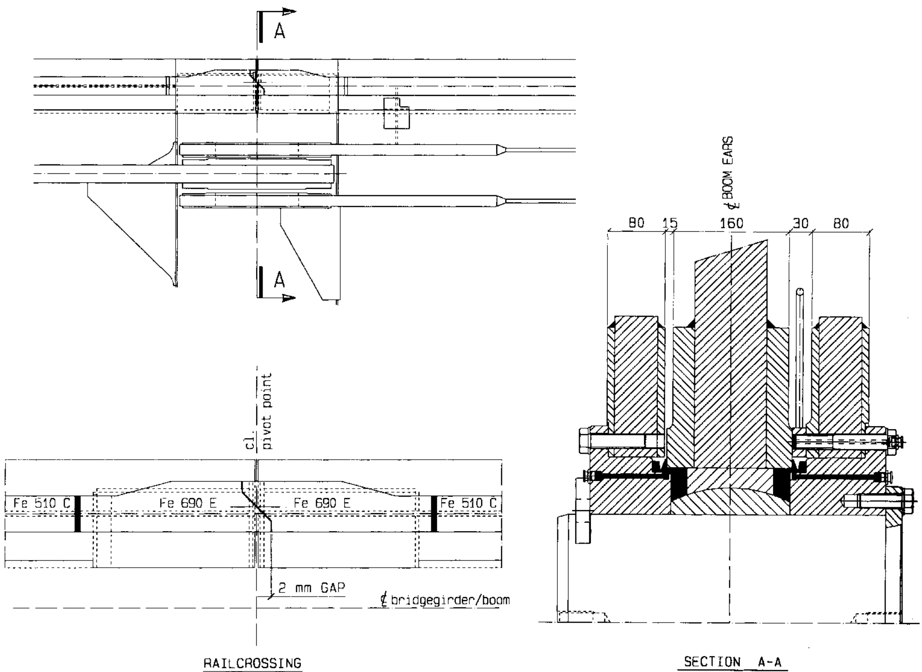


Fig. 8.5.8 Hinge point with widened rail

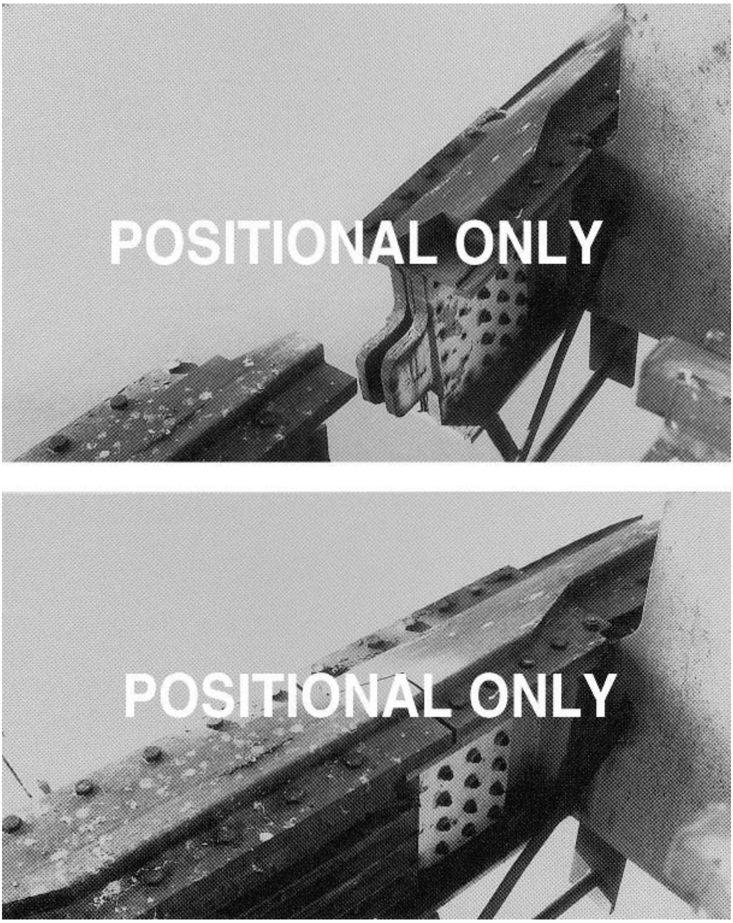


Fig. 8.5.9 Lowering the boom: closing the gap

When the rail is laid on a flexible pad, the pad should be tapered approximately one metre in front of the hinge point. To compensate for this, and to keep the rail level, a correspondingly tapered steel plate section is laid underneath the rail. This tapered section allows the rail, which is somewhat flexible in the vertical plane, to have a transitional trajectory in front of the fixed and non-flexible part of the rail at the boom hinge point. If this is not configured correctly, the rail will fracture at the weakest point. The weak point is directly behind the weld, between the boom hinge point and the section of rail laid on the flexible rail pad.

8.6 Wear and tear of a crane rail

This is a very subjective issue for which a good calculation method cannot be given. Professor Dr Ir. Van Iterson mentioned in 1949 the following manner of comparison of the lifetime of crane rails:

Assume that the allowed wheel load is:

$P = 50 \cdot b \cdot D$ (kg)

Assume that the rails are made of Fe510 and that the wheels are of a better quality. In this case it can be further assumed that the rails are worn out after about one million (10^6) passages of the crane wheel.

For heavier or smaller loads the lifetime decreases or increases with the third degree of the load:

$P_1^3 : P_2^3 = n_2 : n_1$

or

$n_1 \cdot P_1^3 = n_2 : P_2^3$

Whatever we calculate:

– Lining out the wheels correctly is most important to prevent wear and tear of rails and wheels.
– The skew of a crane or trolley is a major cause of wear and tear of rail and wheels.

8.7 Buffers

Cranes are provided with buffers which are intended to cushion the impacts when cranes crash into each other, or into the endstops on the crane tracks. Trollies are provided with buffers which are intended to cushion the impacts when the trolley bumps on to the endstops on the trolley tracks.

Polyurethane elastomer buffers

The micro-cellular structure allows rather high deflections. The material has good resistance against attack by grease, oil, aging and ozone and can be used also in temperatures as low as –15°C. These buffers can provide a deflection of approximately 50 percent of the original height of the buffer.

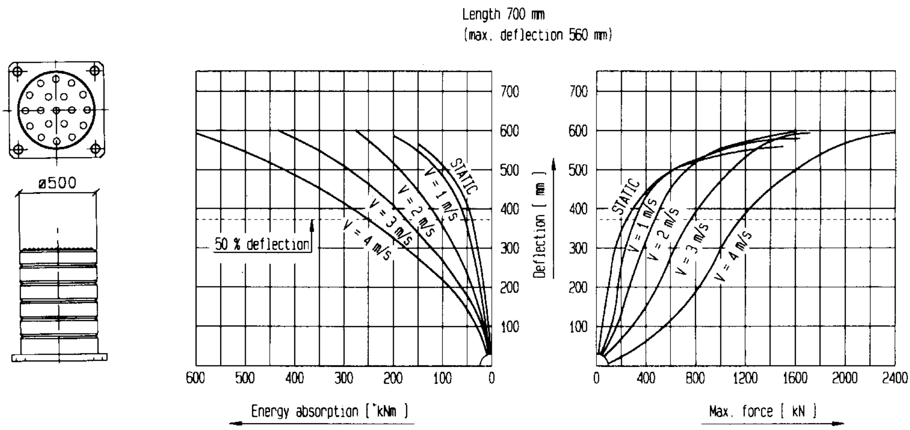


Fig. 8.7.1 Example of a polyurethane buffer

After having calculated the kinetic energy, E, which has to be absorbed, a suitable buffer can be chosen using the available diagrams which demonstrate their properties.

$$\left(E = \frac{Q_t}{2} v^2\,\text{Nm};\ Q_t \text{ in kg};\ v \text{ in m/sec; motors non-driving}\right)$$

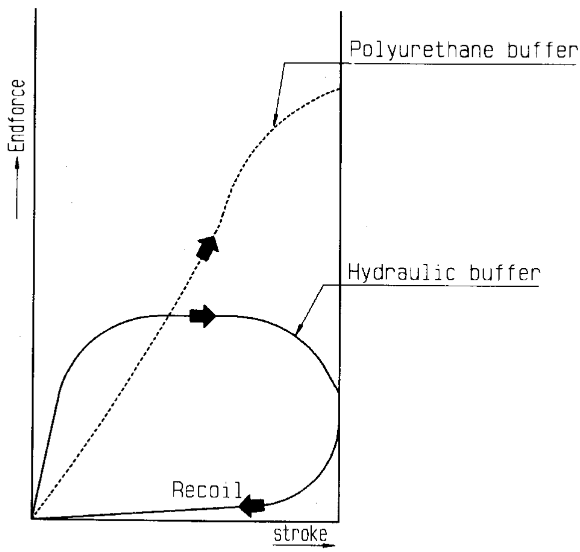


Fig. 8.7.2 Comparison between two buffer types

An example of such a diagram is given for a buffer ϕ 500 mm with a length of 700 mm in Fig. 8.7.1.

Hydraulic buffers

This type of buffer incorporates energy absorption by the displacement of a hollow plunger within the buffer. When the crane or trolley is being driven slowly, the resistance of the buffer is low. This means that the crane or trolley can then use the maximum track length.

Chapter 9

Miscellaneous

9.1 Overload preventers

The main principles concerned are:

1. Overload preventers with strain gauges or load cells.
2. Overload preventers with load measuring pins.

Overload preventers with strain gauges

The strain gauges or load cells can be built-in directly behind a dead-end of a hoisting wire rope or in a yoke which is carrying wire rope sheaves, or underneath a gear-box. Usually the crane driver can check the approximate weight of the carried load on a display in his cabin.

Overload preventers with load measuring pins

High quality stainless steel load pins contain strain gauges which are mounted in a particular way which give a load proportional signal. The load measuring pins can be built-in in a wire rope sheave or in the pin of a hydraulic cylinder. Load monitoring can also be done in the crane drivers cabin etc.

9.2 Snag loads

Occasionally, when a crane driver is joisting a container out of a cell, the container jams because of irregularities in the cell guides. The hoisting winch has to stop in a very short time, as the container snags. In a container crane with a rope trolley, there is a considerable length of

Fig. 9.1.1 Overload preventer

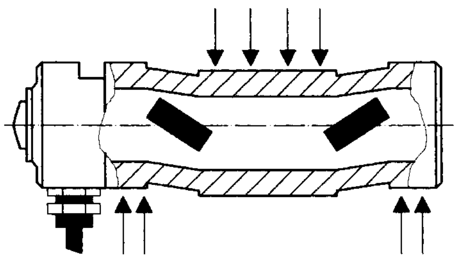


Fig. 9.1.2 Load measuring pin

hoist wire rope, because it runs from the boom end to the trolley, from the trolley to the spreader, and back to the trolley, and from there to the hoisting winch, situated at the rear end of the bridge. This wire rope length will be between approximately 130 and 250 m.

The jamming of the container into the cell, causes an abrupt stop to the hoisting winch and results in a lengthening of the hoisting wire ropes.

$$\Delta l = \frac{F \cdot l}{A \cdot E} \text{cm}$$

where

Δl = elongation of the wire rope in cm
F = the rope pull in kg
l = the total wire rope length as mentioned in cm
A = the *net* area of the cross section of the wire rope in cm^2
E = the elasticity module; say $E = 1{,}000{,}000\ kg/cm^2$ ($10^6\ kg/cm^2$).

The hoist wire rope has a normal safety factor against rupture of about 6. This means that when a considerable wire rope length is stretched by the abrupt halt, the rope elongation, Δl is also considerable.

Under normal speed and load conditions the snag-load will stop the winch without damaging anything, but the safety factor of the wire rope against the rupture will diminish considerably under the snag conditions, e.g. to $\nu = 1{,}5$ or 2.

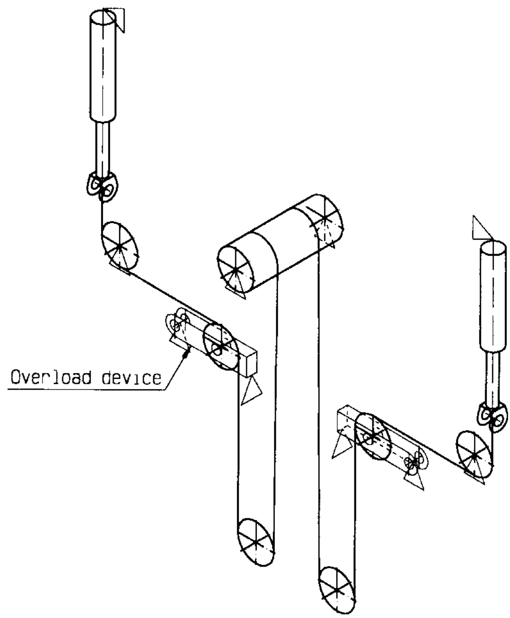


Fig. 9.2.1 Snag load system

With a machinery trolley the wire rope length which may be elongated by an abrupt halt is much shorter. Only 30 to 40 m may be affected when a container jams in a cell directly under the desk of the vessel. It would, therefore, be useful to build in a snag preventer into the construction, thereby preventing problems with container ships with poor cell guides. A fast working snag-damper system can be built in within the machinery trolley comprising 4 snag-dampers each of them with a stroke of approximately 1,5 m and each of them acting on a fixed end of the hoisting wire rope.

The stroke of 1,5 m allows the drum with a diameter of 1,2 m to stop within a rotation of:

$$\frac{1{,}5}{\pi \cdot 1{,}2} \cdot 360 \text{ degrees} = 143 \text{ degrees or}$$

over a distance measured over the circumference of the drum of also 1,5 m.

The hoisting winch may for example have motors with a maximum torque of $f_a = 1{,}6$. The snag damper system must then give way when $M \geq 1{,}6$ Mn, let us say at $f_a = 1{,}8$ and immediately the motor current must then be brought to zero. An emergency stop must be commenced to stop the winch.

If there is a 'concentric snag', all four wire ropes will be overloaded. If there is an 'eccentric snag', only two wire ropes are overloaded. These wire ropes have to stop the winch in the same stroke. The overload in these ropes under eccentric snag conditions will be twice as bad as under concentric snag conditions.

If the crane driver wants to hoist with an empty spreader at higher speeds (field-weakening speeds) and the spreader jams into the cell, the overload in the wire ropes becomes still higher, due to the higher motor speeds, which means a longer braking time.

Example

Machinery trolley Scheme (Fig. 9.2.1)	Full loaded trolley	Trolley with empty container
Hoisting speed of the load within the cells of the ship:		
v(m/min)	$v = 60$ m/min	$v = 90$ m/min
Wire rope speed on the drum:		
$v_1 = (2v : 60)$ (m/sec)	$v_1 = 2$ m/sec	$v_1 = 3$ m/sec
Hoisting capacity on the ropes: Q (tons)	$Q = 66$ t	$Q = 18$ t

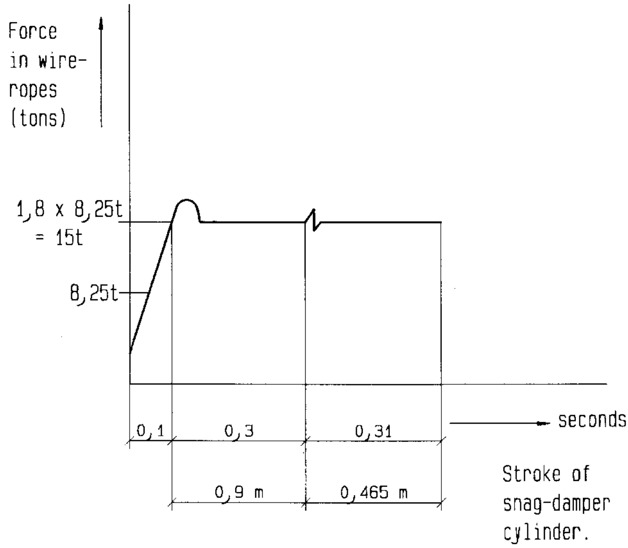


Fig. 9.2.2 Snag load forces in system according to Fig. 9.2.1

Machinery trolley Scheme (Fig. 9.2.1)	Full loaded trolley	Trolley with empty container
Snag	Eccentric	Eccentric
Nos. of revolutions of the hoisting motor(s)	$n = 783$ rev/min	$n = 1175$ rev/min
$\omega = (n_{motor} : 60) \cdot 2\pi$ rad/sec	$\omega = 81{,}95$ rad/sec	$\omega = 123$ rad/sec
Inertia movement on motorshaft from motor(s), brake sheaves and gear box:		
J_{rot} (kg m^2)	$J_{rot} = 46$ kg m^2	$J_{rot} = 46$ kg m^2
Brakes will be in action after Δ_t sec	$\Delta_t = 0{,}3$ sec	$\Delta_t = 0{,}3$ sec
Effective brake movement:		
M_b (Nm)	$M_b = 18\ 050$ Nm	$M_b = 18\ 050$ Nm
Snag damper system gives way when $f_a = 1{,}8$. The motor current is then immediately switched to zero		

Machinery trolley Scheme (Fig. 9.2.1)	Full loaded trolley	Trolley with empty container
In $\Delta_t = 0{,}3$ sec the wire rope on the drum with diameter $D = 1{,}2$ m travels over the circumference of the drum over the 'overrun stroke' of:		
$S_1 = \Delta_t \cdot v_1$ (m)	$S_1 = 0{,}3 \cdot 2 = 0{,}6$ m	$S_1 = 0{,}3 \cdot 3 = 0{,}9$ m
Attention: The snag damper system must be able to follow the stroke S in the available – very short time in order to prevent over-stretching the wire ropes!		
During mechanical braking the effective braking time becomes:		
$t_{\text{brake}} = \dfrac{\omega \cdot J_{\text{rot}}}{M_b}$ (sec)	$t_b = \dfrac{81{,}95 \cdot 46}{18\,050}$ $t_b = 0{,}21$ (sec)	$t_b = \dfrac{123 \cdot 46}{18\,050}$ $t_b = 0{,}31$ sec
During t_b sec, the wire rope on the drum travels over the circumference of the drum over:		
$S_2 = \frac{1}{2} \cdot v_1 \cdot t_b$ (m)	$S_2 = \frac{1}{2} \cdot 2 \cdot 0{,}21$ $= 0{,}21$ m	$S_2 = \frac{1}{2} \cdot 3 \cdot 0{,}31$ $= 0{,}465$ m
The total absorbed wire rope length is:		
$S_t = S_1 + S_2$ (m)	$S_t = 0{,}6 + 0{,}21$ $= 0{,}81$ m	$S_t = 0{,}9 + 0{,}465$ $= 1{,}365$ m
The stroke of the snag damper is:		
S_d (m)	$S_d = 1{,}5$ m	$S_d = 1{,}5$ m
Result: The wire ropes are not overstretched any more, so the wire rope safety factor is not reduced further than $v = \dfrac{6}{1{,}8} =$ approximately 3		

If immediately after tripping the motor current is reversed, thereby aiding the braking torque, the stroke of the snag damper may be marginally reduced.

Note: If extra brakes are installed or if caliper brakes (with a reaction time of approximately 0,1 sec instead of 0,3 sec) are installed on the flange of the wire rope drums, the braking time and the 'overrun stroke' can be reduced still further.

If a snag device is not incorporated into the hoisting mechanism, there are quite different rope pull forces involved during a snag.

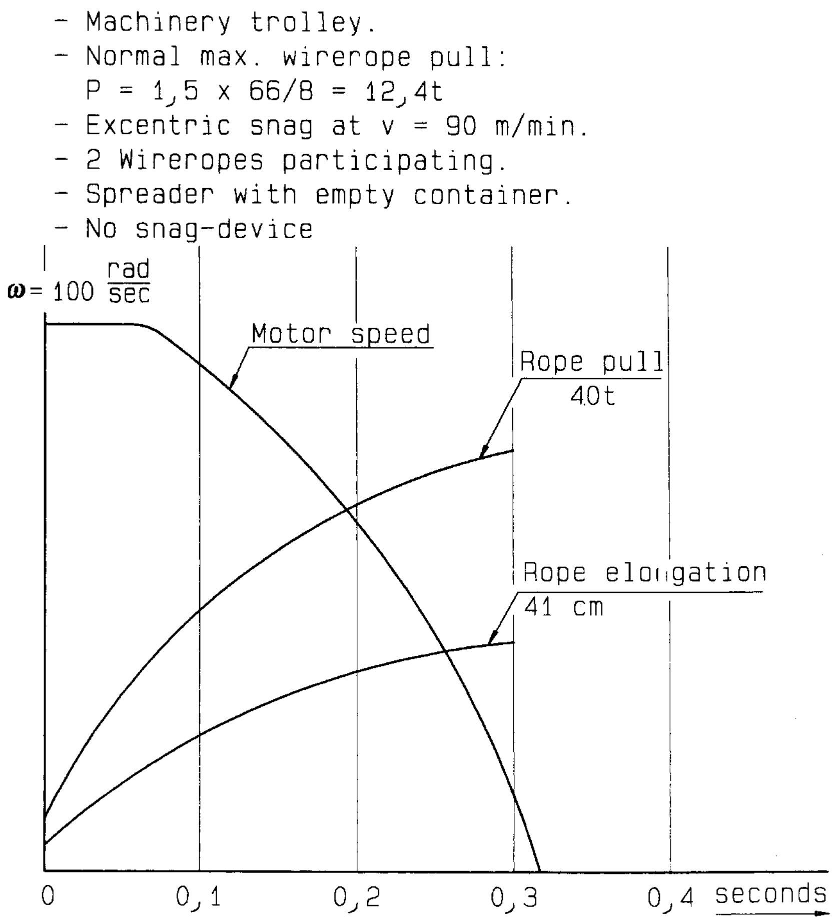


Fig. 9.2.3 Snag in hoisting winch of machinery trolley

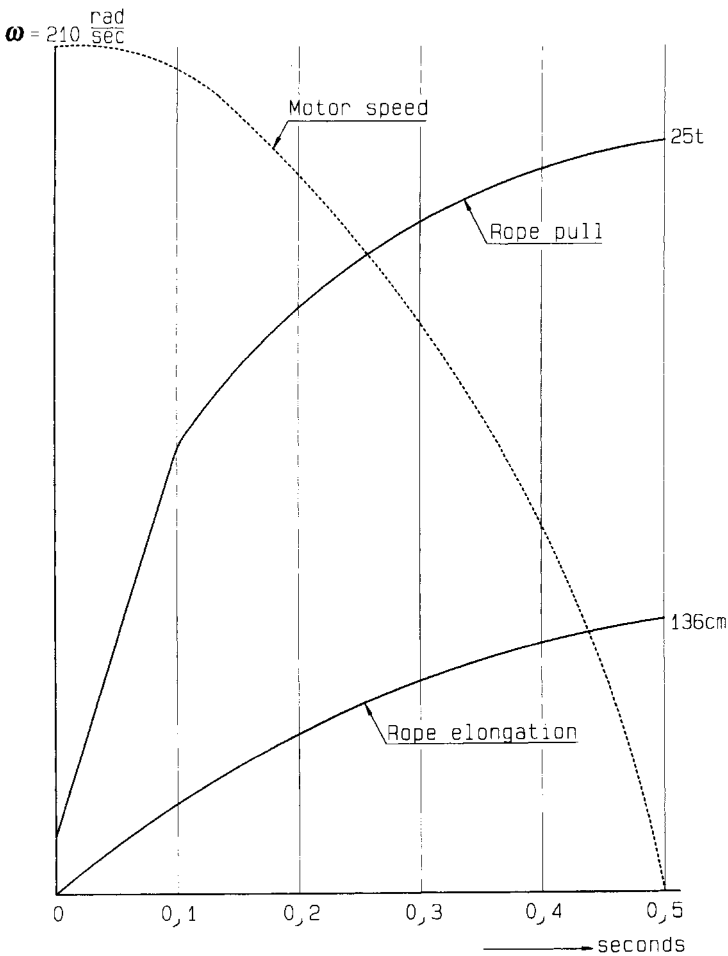


Fig. 9.2.4 Snag in hoisting winch of full-rope trolley:

- **Eccentric snag at *v*=170 m/min**
- **3 wire ropes participating**
- **Empty spreader**
- **No snag device**

9.3 Anti-collision systems

Some anti-collision systems work using the principles of:

– sonar;
– radar;
– low frequency near-field induction.

Sonar

Although the principle is very good, there is the danger with this system that a strong wind can blow the sonar waves away.

Radar

Radar usually uses advanced microwave or Doppler radar technology combined with some digital signal processing. Safety circuits are built-in, and the system can sense objects up to a distance of about 40 m. The maximum crane travel speed is about 200 m/min.

The radar beam which is sent out will be reflected by solid objects and will be received by the same radar antenna. It can measure the distance between two objects and also the speed with which the object is approaching the sensed object.

Low-frequency near-field induction system

These low frequency systems work at a frequency of approximately 90 to 220 kHz and have a working maximum range of about 30 m. A transmitter and antenna is installed on the first crane and a receiver and antenna on the adjacent crane. With this system it is possible to install three distance steps between the cranes, which should be respected. For example:

- at 30 m distance an audible signal is given;
- at 20 m distance the crane speed is decreased;
- at 5 m distance the final stop signal is given.

9.4 Cable reels

The current supply to a crane is normally delivered by a motor-driven supply reel on which a tubular MV cable or sometimes a flat MV cable is wound. The tension can vary from 3,3 kV to 23 kV; fibreoptic cores can be included in the current supply cable. These fibreoptic cores need a special fibreoptic rotary accumulator which is vulnerable to damage.

The cable reels can be:

- parallel type or drum type;
- monospiral radial type;
- random lay radial type;
- the pull and store type.

Heat dissipation has to be controlled and this will normally give a de-rating factor which must be applied to the current capacity of the cable.

Manufacturers use all types of drive. A difficult issue can be the turn-over point, if a fast running crane, with a cable reel with a big inertia moment has to run at full speed over the turn-over point. At this turn-over point, the fully loaded cable reel, running at full speed in one direction, has to go immediately in the opposite direction. A special system is the pull and store system in which the cable tension is limited by the fact that the pull reel lifts up the cable on a constant reeling radius. Using this configuration, it is possible to limit the maximum tension of the cable within the allowed limits without the need for a sophisticated torque control.

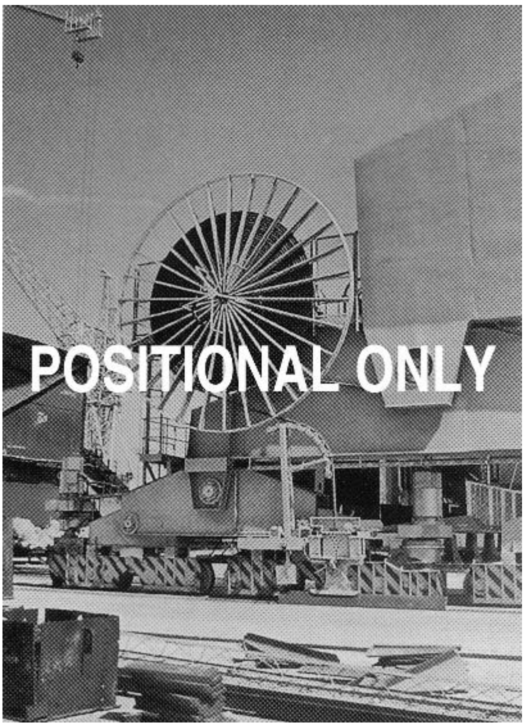


Fig. 9.4.1 Giant monospiral radial type cable reel

9.5 Festoon systems: current and data supply to trolleys

The current and data supply between the fixed part of the crane and the travelling trolley is normally carried out by a festoon system. In a crane with a machinery trolley, the festoon system incorporates a large

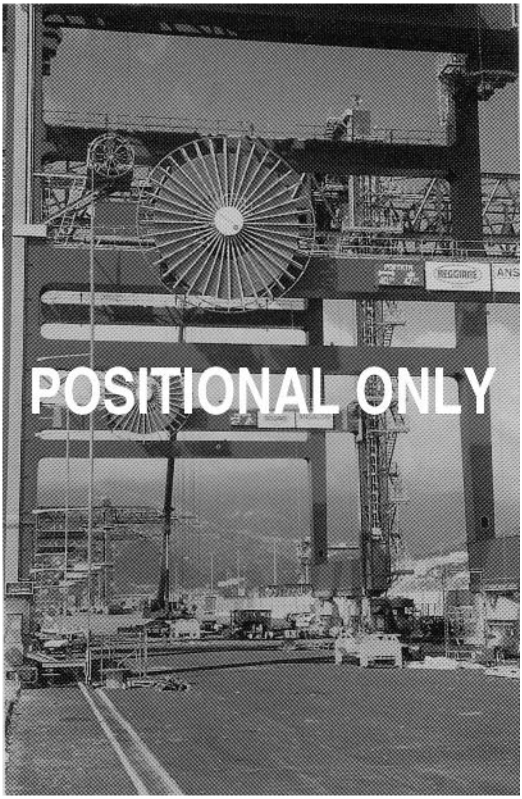


Fig. 9.4.2 Pull and store cable reel

number of rather heavy current supply cables. Cranes with full rope trolleys, may even have as few as 2 or 3 thin cables. The number of these cables has been sharply reduced over the years, by the use of fibreoptic connections, which are now frequently used. The cables in the festoon system are mounted on cable carriers and can be tubular or flat cables. In the fast moving machinery trolleys, the use of 2 or 3 motor-driven festoon carriers is necessary to move the big bundles or heavy festoon cable more smoothly. Alternatively, current supply rails can be used. When these are of the normal copperhead type, double current collectors are required.

The wear of such systems can be considerable when high amperages have to be transferred and crane or trolleys are moving at high speed. The data supply could then be carried out via a guided microwave data communication system.

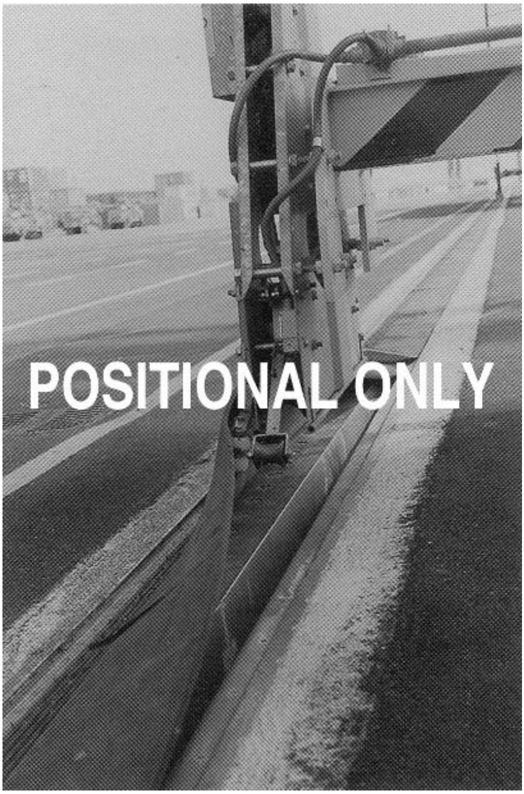


Fig. 9.4.3 Covering the current supply cable channel

Unipole insulated conductor rails can also be used for transporting the main current to a crane or a machinery trolley. These conductor rails can have a copper cross-sectional area of 500 mm^2 and may have a continuous Ampère capacity of approximately 1000 Amps. The required capacity should be carefully calculated, as well as the voltage drop over the distance from the point where the system is fed to the end of the crane or trolley travelling length.

9.6 Inductive power transfer and data transmission

Based on the developments of the University of New Zealand in 1990, Wampfler started in 1996 with further research and development of 'contactless festoon systems for cranes and people movers'. Nowadays a lot of experience has been gained on these systems in the automobile industry, where many contactless, power supplied floor conveying

Fig. 9.4.4 Cable reels on an unloader

Fig. 9.5.1 Current supply to machinery trolley (including driven cable carrier)

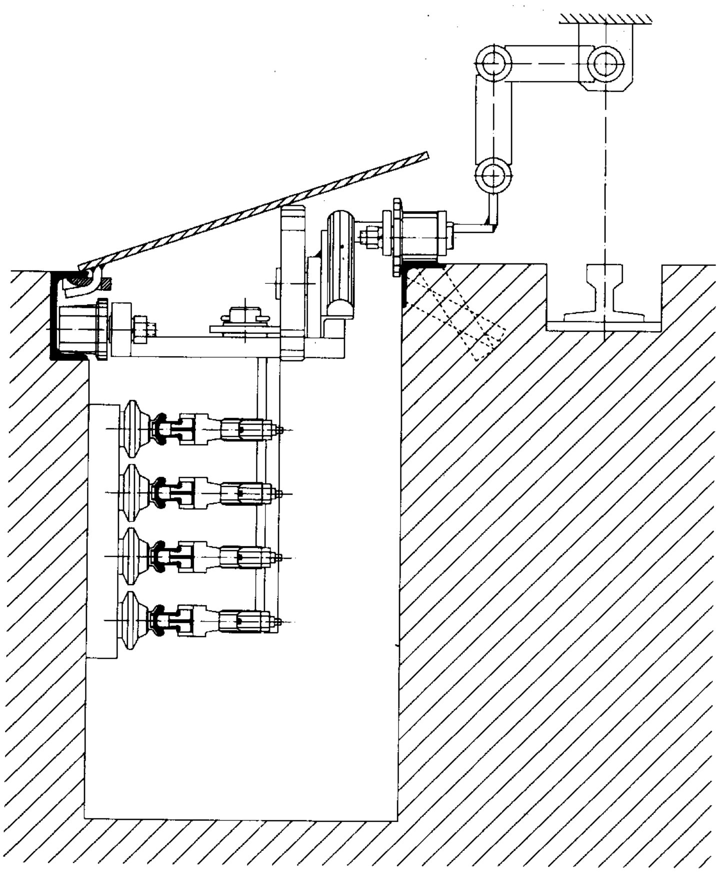


Fig. 9.5.2 Current supply rails within a trench

systems are used, experience has also been gained in the people-movers industry.

In the crane industry the use of these systems is still limited, although a progressive process will show interesting results in the coming decade.

Compared with the normal festoon systems, as described in Section 9.5, the non-contact festoon systems have many advantages, among others:

- there is no speed and acceleration limit;
- wind and storm have no influences on these packages;
- it is non-microwave and non-RF;
- eventual damages can easily be repaired;
- maintenance is minor and simple;
- the system is EMC-proof.

IPT Power Transfer

This fully patented system is now available for a power requirement up to 160 kW. Trials have yet to be done with a system of 240 kW and higher. The system is based on the principle of magnetic induction in the frequency of approximately 20 kHz (VLF, very low frequency) which does not present a health hazard. Figure 9.6.1 shows this unit.

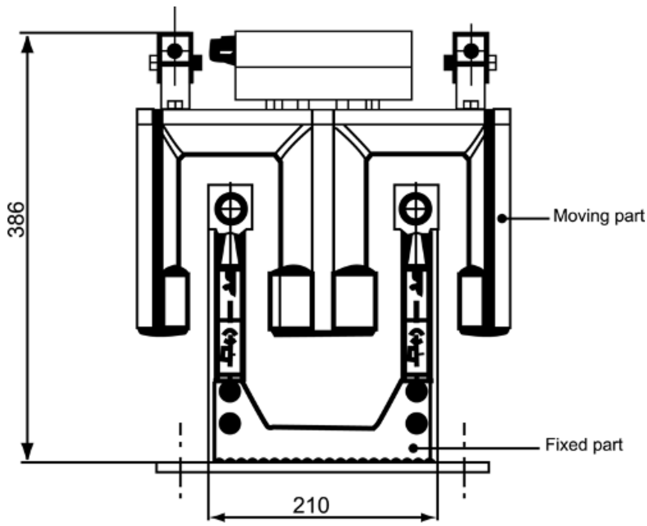


Fig. 9.6.1 IPT power transfer arrangement

The fully patented data transmission system Powertrans RII is an interference-proof telemetric system for the transfer of data-, video- and audio-signals. The system operates in full-duplex transmission and achieves a data rate up to 4 Mbits/sec; (illustrated in Fig. 9.6.2).

The system uses an ISM band and DSSS (Direct Sequence Spread Spectrum); it is bugproof and is not susceptible to interference. It can be used with contactless IPT power supply systems.

Practical solutions

In the power transfer on cranes with a luffable boom, the connection of the IPT system between the bridge and the boom must be carefully designed, as the tolerances between the IPT system and the pick-ups are rather narrow, typically ±25 mm. A doubling of the pick-up is not needed to cover the gap between the bridge and the boom system.

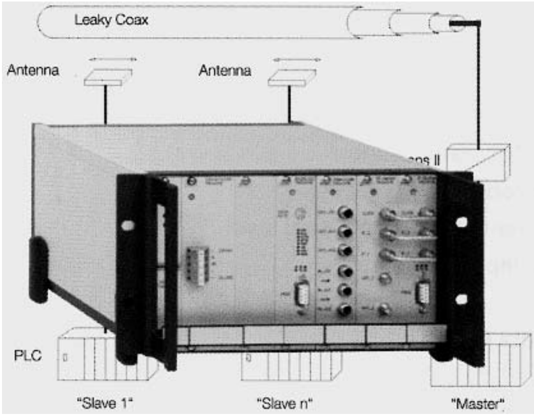


Fig. 9.6.2 Data transmission Powertrans® II

9.7 Hoppers

In the bulk unloaders (see Fig. 1.4.2) the grab is unloaded into a hopper. The wider the top-opening of the hopper, the quicker the incoming trolley can unload the grab, and the less material is lost during the unloading. The walls of the hopper should not have a slope of less than approximately 60 degrees to the horizontal. The edges between the back

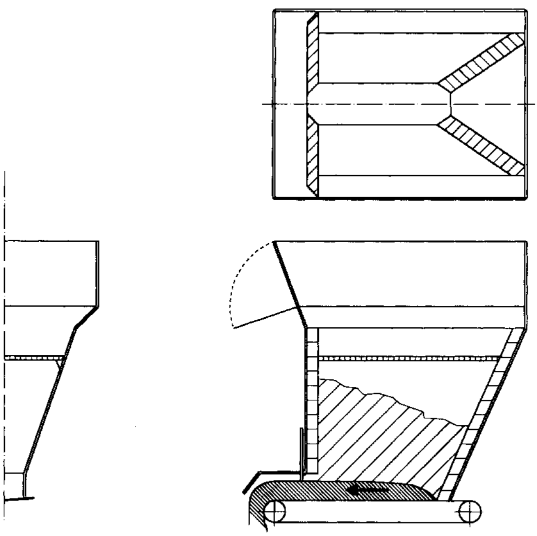


Fig. 9.7.1 Hopper with conveyor

plate of the hopper and the side plates, should be flattened out in order to prevent the clogging of the material in the hopper in sharp corners.

It is strongly advisable to build a sturdy grid in the top of the hopper as it cushions the shocks when the material is dropped into the hopper by the grab. It also catches lumps, wood, and all other rogue material and prevents clogging in the chutes of the conveyor-belt system in the unloader and behind the unloader. The walls of the hopper can wear out rather quickly through the abrasion, particularly by wet material. Abrasion resistant plates can be built-in to prevent this wear as far as possible. If required a dust-suppression system can be built-in as well as vibrators to prevent clogging of the material.

9.8 Apron feeders; conveyors

To unload the hopper a number of mechanisms can be used, for example:

- apron feeders (for ore and similar materials);
- heavy duty belt conveyors (for coal);
- vibrating feeders.

Each mechanism has its own advantages and disadvantages. Apron feeders and conveyors in an unloader can only offer a very limited distance for transportation. The heavy load from the material in the fully-loaded hopper resting on the feeder or belt, and the extra force which is needed to draw the material out of the hopper, require considerable power.

Problems with hoppers and conveyors can be overcome as follows:

- A wide feeder with enough body to take up the impact of the material in the hopper should be used.
- A grid in the top of the hopper should be fitted to cushion the shocks of the material dropping out of the grab opened above the hopper.
- Construction of skirts between the hopper and the conveyor must be very carefully engineered in order to prevent damage to the conveyor or apron feeder.
- Enough power must be available to drive the conveyor and apron feeder. The brake-out-force from the material out of the hopper can be considerable.

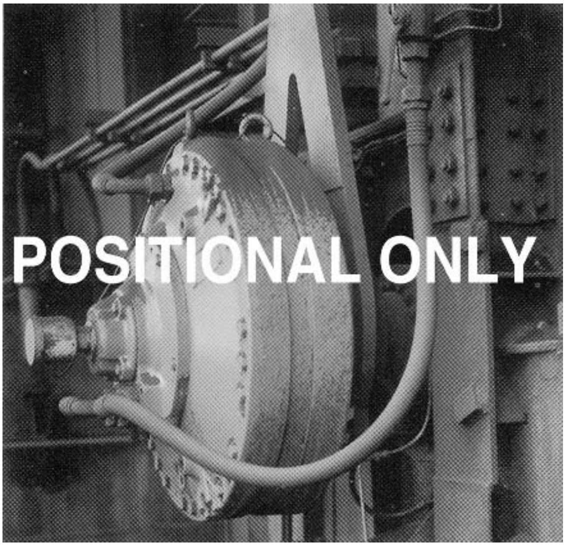


Fig. 9.8.1 Hydraulic drive for conveyor underneath a hopper

An approximate method of calculation of the required horsepower for a conveyor underneath a hopper is given.

Main characteristics

Capacity: $Q = 2500$ t/hr

Material: Coal, density $d = 1$ t/m^3

Conveyor speed: $v = 0{,}6$ m/sec

Length of the loaded part of the conveyor: $L = 8$ m

Opening width of the hopper: $B = 1{,}8$ m

Height of the load on the conveyor:

$$H = \frac{Q}{B \cdot d \cdot v \cdot 3600} \text{ m}$$

$$H = \frac{2500}{1{,}8 \cdot 1 \cdot 0{,}6 \cdot 3600}$$

$$H = 0{,}64 \text{ m}$$

1. Resistance by the hopper and conveyor skirts (kW):

$$N_1 = \frac{2 \cdot \mu \cdot L \cdot d \cdot H^2 \cdot v \cdot 10^3}{102 \cdot \eta}$$

$$N_1 = \frac{2 \cdot 1 \cdot 8 \cdot 1 \cdot 0{,}64^2 \cdot 0{,}6 \cdot 10^3}{102 \cdot 0{,}9}$$

$$N_1 = 43\,\mathrm{kW}$$

2. Resistance through 'drawing out the material' out of the full loaded hopper (kW):

$$N_2 = \frac{\mu^1 \cdot B \cdot L^1 \cdot v \cdot 10^3}{102 \cdot \eta}$$

$$N_2 = \frac{0{,}5 \cdot 1{,}8 \cdot 6{,}5 \cdot 0{,}6 \cdot 10^3}{102 \cdot 0{,}9}$$

$$N_2 = 38\,\mathrm{kW}$$

3. Resistance through the moving conveyor itself (kW):

$$N_3 = \frac{f \cdot G_\mathrm{m} \cdot L^{11} \cdot v}{102 \cdot \eta}$$

where:

f = resistance coeff. of the moving conveyor: $f = 0{,}65$

G_m = weight of the moving part per metre of the upper and under strand: $G_\mathrm{m} = 240$ kg/m

L^{11} = centre to centre length of the end drums: $L^{11} = 9{,}5$ m

v = conveyor speed: $v = 0{,}6$ m/sec

η = total efficiency: $\eta = 0{,}9$

$$N_3 = \frac{0{,}65 \cdot 240 \cdot 9{,}5 \cdot 0{,}6}{102 \cdot 0{,}9}$$

$$N_3 = 10\,\mathrm{kW}$$

The necessary motorpower to drive the conveyor is:

$$N = N_1 + N_2 + N_3\ \mathrm{kW}$$
$$N = 43 + 38 + 10 = 91\ \mathrm{kW}$$

9.9 Electronic Tracking Guide System

C E Plus GmbH of Magdeburg in Germany have developed a fully patented Electronic Tracking Guide System, which can be used for the crane travelling mechanisms of overhead cranes, bridge cranes and similar equipment. These cranes must be equipped with separate AC frequency units for the travelling mechanisms on each rail. The system works as follows:

- On one rail two sensors are built-in, in front of the outer wheels. (See Fig. 9.9.1.)
- These sensors measure, without contact, the distance between the flange of the wheel and the side of the crane rail.
- If a difference between the measurements of the sensors arises during crane travelling, the controlling computer between the drives, commands one of the frequency converters to speed-up or to slow-down the travelling a little. The speed of one side of the crane is thus regulated in such a way that the distance between the rail and the wheelflange is corrected. In this way the skidding of the crane is prevented.

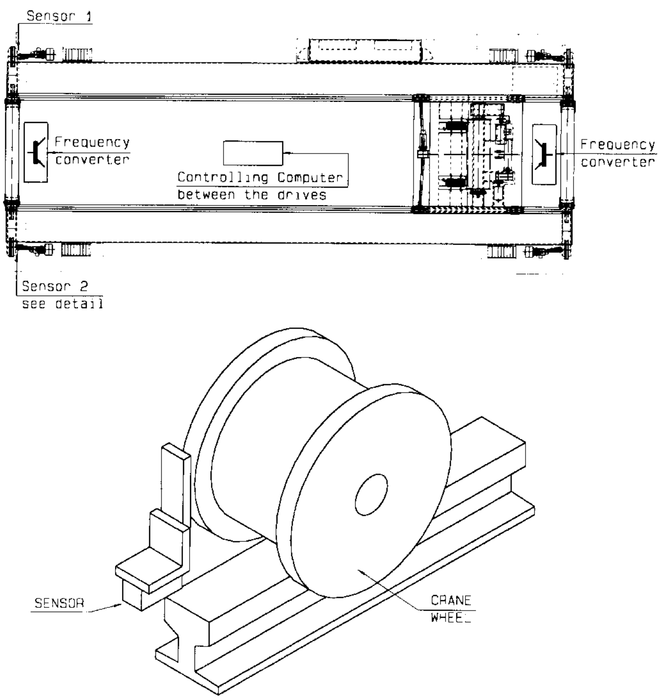


Fig. 9.9.1 The electronic tracking guide system

The two sensors control these distances continuously and in doing so prevent the flanges of the wheels from hitting the side of the crane rail. Therefore, wear and tear of the wheel flanges and the crane rail can be minimized. In order to give the system enough time to react, the free space between the rail and the wheelflanges, which is normally 12–15 mm, should be increased to 30 to 50 mm. Experience with a 35 ton overhead crane with a maximum travelling speed of 100 m/min has shown that the system works well.

9.10 Gears

There are a number of sophisticated computer programs with which gearings can be calculated. ISO, DIN and AGMA have – among others – comprehensive calculations for all sorts of gearing arrangements.

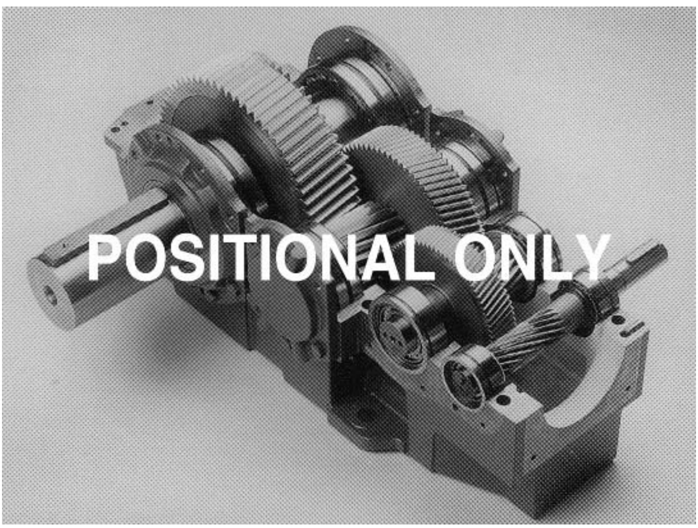


Fig. 9.10.1 Gearbox for a hoisting mechanism

In this section only a rough calculation method for helical gears is given, which can be used to get an idea of the way in which gears should be dimensioned and calculated. This calculation method is derived from A. K. Thomas, Wissmann, Niemann and Verschoof.

For a more comprehensive explanation of this complicated subject, please study the ISO, DIN or AGMA calculations.

Calculation on fatigue (pitting)

$$N_{\text{all}} = k_1 \cdot b \cdot d_n^2 \cdot y_1 \cdot Q_w \cdot \frac{n}{2 \cdot 10^5}\ \text{kW}$$

where

– Power which can be transmitted:	N_{all}	in kW
– Modulus:	m	in cm
– Number of teeth:	z_1 z_2	pinion wheel
– Pitch diameter:	d_0	$d = m \cdot z$ (cm)
– Helix angle:	β	degrees
– Addendum coefficient:	x_1 x_2	$x_1 = + \cdots \cdot m$ (cm) $x_2 = + \cdots \cdot m$ (cm)
– Centre distance:	a	cm $a = \dfrac{(z_1 + z_2) \cdot m}{2 \cos \beta} + \dfrac{(x_1 + x_2) \cdot m}{\cos \beta}$ cm
– Allowable strength for pitting:	k_1	$k_1 = 360\ \text{kg/cm}^2$ for 17 CrNiMo6 $k_1 = 60\ \text{kg/cm}^2$ for C60N
– b = width of pinion and wheel:	b_1 b_2	in cm in cm
– d_n = theoretical pitch diameter:	d_n	in cm $d_n = \dfrac{m \cdot z + 2 \cdot x \cdot m}{\cos^2 \beta}$

– y = Coefficient: y_1 y_1 for pinion
y_2 y_2 for wheel

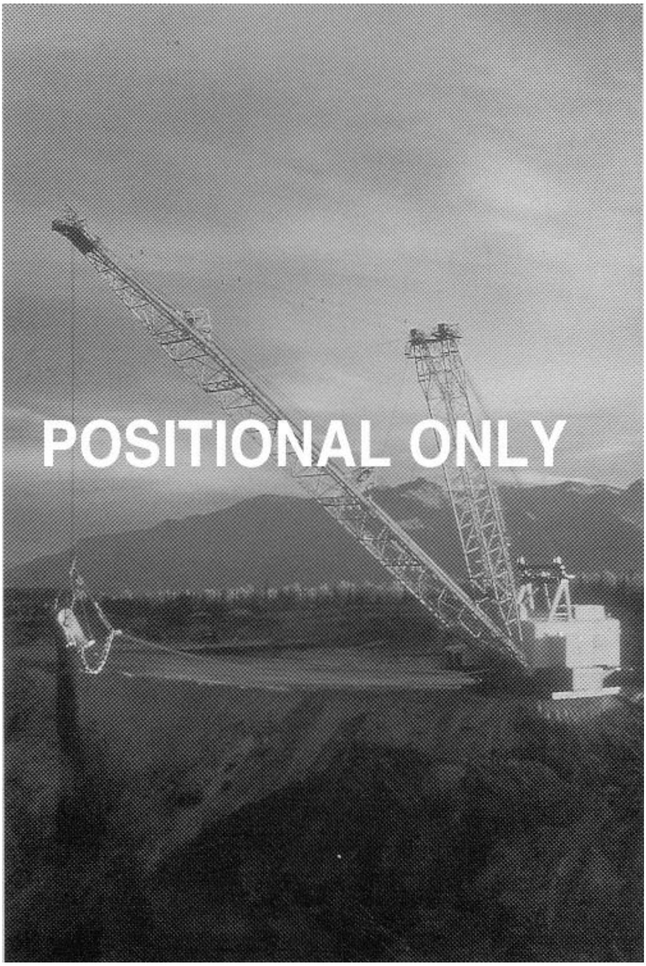


Fig. 9.10.2 Bucyrus-Erie 1300 walking dragline

	y_1 for $i=$				y_2 for $i=$			
Z_1	1	2	5	∞	1	2	5	∞
10	–	0,122	0,135	0,148	–	0,240	0,311	0,329
12	0,098	0,148	0,167	0,187	0,098	0,236	0,301	0,327
14	0,142	0,165	0,190	0,216	0,142	0,233	0,294	0,327
17	0,150	0,180	0,212	0,243	0,150	0,229	0,287	0,326
20	0,155	0,189	0,225	0,260	0,155	0,226	0,282	0,325
30	0,159	0,201	0,246	0,289	0,159	0,221	0,275	0,324
50	0,160	0,207	0,256	0,304	0,160	0,217	0,272	0,323
150	0,162	0,213	0,265	0,318	0,162	0,215	0,265	0,322

– Q_w = coefficient, related to helix angle β

Q_w =	1,0	1,11	1,22	1,31	1,40	1,47	1,54	1,60	1,66	1,71
β_0 =	0°	5°	10°	15°	20°	25°	30°	35°	40°	45°

– Number of revolutions: n_1 for pinion, in rev/min
n_2 for wheel, in rev/min

$$N_{all} = k_1 \cdot b \cdot d_n^2 \cdot y_1 \cdot Q_w \cdot \frac{n}{2 \cdot 10^5} \text{ kW}$$

– Power to be transmitted: N (with . . . % rating)

– $f_p = \dfrac{N_{all}}{N}$

$f_p = 2{,}2 = 50\ 000$ hours

$f_p = 1{,}75 = 25\ 000$ hours

$f_p = 1{,}40 = 12\ 500$ hours

Calculation on strength

$$\sigma_b = \frac{F \cdot q}{b \cdot m \cdot e \cdot Q} \text{ kg/cm}^2$$

where:

– σ_b = bending stress: σ_b in kg/cm^2
– Power to be transmitted: N in kW
– Nominal force on the teeth: F in kg

$$F = \frac{N \cdot 95\,500}{n \cdot d/2} \text{ kg}$$

– Number of revolutions: n in rev/min
– Nominal pitch diameter: d $d = m \cdot z$ (cm)
– Modulus: m in cm
– Multiplication coefficient depending on addenum, $X_{1,2}$: q
– b = width of pinion and wheel: b_1 in cm
b_2
– Coefficient depending on the numbers of the mating teeth: e
– Coefficient, depending on the helix angle β Q

– Multiplication coefficient q,
depending on X_1, X_2

	Z =	10	12	15	20	25	30	40	60	90	150
—	+ 0,5	q = 2,9	2,8	2,7	2,6	2,55	2,5	2,45	2,4	2,35	2,3
—	+ 0,4	q = 3, –	2,9	2,8	2,65	2,6	2,55	2,5	2,45	2,4	2,35
—	+ 0,3	q = 3,3	3,1	2,9	2,75	2,7	2,6	2,55	2,5	2,45	2,4
X =	+ 0,2	q = 3,7	3,4	3,2	2,9	2,8	2,75	2,65	2,6	2,45	2,4
—	+ 0,1	q = 4,3	3,9	3,5	3,1	3, –	2,9	2,8	2,65	2,5	2,45
—	+ 0, –	q =	4,5	3,9	3,4	3,2	3,1	2,9	2,7	2,55	2,5
—	– 0,1	q =		4,5	3,8	3,4	3,3	3, –	2,8	2,6	2,55
—	– 0,2	q =			4,2	3,7	3,4	3,2	2,95	2,7	2,6
—	– 0,3	q =				4,1	3,65	3,35	3,1	2,8	2,7
—	– 0,4	q =				4,4	3,9	3,5	3,2	2,9	2,75
—	– 0,5	q =					4,1	3,7	3,35	3, –	2,85

– Coefficient e:

Nos. of teeth Z_1		12	14	18	28	50	100
Nos. of	12	1,25	1,25	1,25	1,35	1,45	1,50
teeth Z_2	18	1,30	1,30	1,30	1,45	1,50	1,55
	50	1,30	1,35	1,35	1,50	1,60	1,65
	100	1,30	1,35	1,40	1,55	1,65	1,70
	∞	1,30	1,35	1,45	1,60	1,70	1,75

– Coefficient Q:

Q =	1,0	1,2	1,28	1,33	1,35	1,36
β =	0°	5°	10°	15°	20°	25°

$$\sigma_{\max} = \frac{F \cdot q}{b \cdot m \cdot e \cdot Q} \text{ kg/cm}^2$$

σ_{allowed} $\bar{\sigma}$

$\bar{\sigma} = 2000$ kg/cm^2
for 17 CrNiMo6

$\bar{\sigma} = 1600$ kg/cm^2
for C60N

$$f_s = \frac{\sigma_{\text{all}}}{\sigma_{\max}}$$

Example

Gearbox for hoisting mechanism	1st stage	2nd stage	3rd stage
Power to be transmitted	800 kW	784 kW	768 kW
Gear ratio	2,35	2,68	3,05
Total gear ratio		$2{,}35 \cdot 2{,}68 \cdot 3{,}05 = 19{,}20$	
Numbers of rev/min of pinion	800 r/min	340,4 r/min	127 r/min
Numbers of rev/min of wheel	(340,4 r/min)	(127 r/min)	(41,6 r/min)
Centre distance a	28,0 cm	40,0 cm	63,0 cm
Numbers of teeth of pinion	$Z_1 = 20$	$Z_1 = 19$	$Z_1 = 19$
Numbers of teeth of wheel	$Z_2 = 47$	$Z_2 = 51$	$Z_2 = 58$
Normal module	0,8 cm	1,1 cm	1,6 cm
Face width of pinion	14,0 cm	20,0 cm	28,0 cm
Face width of wheel	13,6 cm	19,6 cm	27,6 cm
Addendum coefficient of pinion	+ 0,5	+ 0,3	+ 0,2
Addendum coefficient of wheel	+ 0,3	+ 0,3	− 0,2

Helix angle $\beta°$	12°	12°	12°
Cosinus $\beta°$	0,9781	0,9781	0,9781
Material of the pinion	17CrNiMo6 Hardened and ground	17CrNiMo6 Hardened and ground	17CrNiMo6 Hardened and ground
Material of the wheel	Same	Same	Same
Control of centre distance a	$a = \frac{(20+47) \cdot 0{,}8}{2 \cdot 0{,}9781} + \frac{(0{,}5+0{,}3) \cdot 0{,}8}{0{,}9781}$ $= 27{,}40 + 0{,}65$ $= 28$ cm	$a = \frac{(19+51) \cdot 1{,}1}{2 \cdot 0{,}9781} + \frac{(0{,}3+0{,}3) \cdot 1{,}1}{0{,}9781}$ $= 39{,}36 + 0{,}67$ $= 40$ cm	$a = \frac{(19+58) \cdot 1{,}6}{2 \cdot 0{,}9781} + \frac{(0{,}2-0{,}2) \cdot 1{,}6}{0{,}9781}$ $= 62{,}98 + 0$ $= 63$ cm
k_1	360 kg/cm^2	360 kg/cm^2	360 kg/cm^2
b_1(= bearing width of b_2)	$b_1 = 13{,}6$ cm	$b_1 = 19{,}6$ cm	$b_1 = 27{,}6$ cm
$d_n = \frac{m \cdot z + 2 \cdot x \cdot m}{\cos^2 \beta}$	$= \frac{0{,}8 \cdot 20 + 2 \cdot 0{,}5 \cdot 0{,}8}{0{,}9781^2}$ $= 17{,}27$ cm	$= \frac{1{,}1 \cdot 19 + 2 \cdot 0{,}3 \cdot 1{,}1}{0{,}9781^2}$ $= 22{,}53$ cm	$= \frac{1{,}6 \cdot 19 + 2 \cdot 0{,}2 \cdot 1{,}6}{0{,}9781^2}$ $= 33{,}44$ cm
y_1	$y_1 = 0{,}19$	$y_1 = 0{,}19$	$y_1 = 0{,}191$
Q_w	$Q_w = 1{,}26$	$Q_w = 1{,}26$	$Q_w = 1{,}26$
n_1	$n_1 = 800$ r/min	$n_1 = 340{,}4$ r/min	$n_1 = 127$ r/min

$$N_{all} = k_1 \cdot b \cdot d_n^2 \cdot y_1 \cdot Q_w \cdot \frac{n}{2 \cdot 10^5}\,\text{kW}$$

N_{all}	$N_{all} = 1397$ kW	$N_{all} = 1459$ kW	$N_{all} = 1597$ kW
N	$N = 800$ kW	$N = 784$ kW	$N = 768$ kW

Pitting

$f_p = N_{all} : N$	$f_p = 1397 : 800$ $= 1{,}75$	$f_p = 1459 : 784$ $= 1{,}86$	$f_p = 1597 : 768$ $= 2{,}08$

Strength

$$\sigma_b = \frac{F \cdot q}{b \cdot m \cdot e \cdot Q} \text{ kg/cm}^2$$

$F = \dfrac{N \cdot 95\,500}{n \cdot (m \cdot z)/2}$	$F = \dfrac{800 \cdot 95\,500}{800 \cdot (0{,}8 \cdot 20)/2}$	$F = \dfrac{784 \cdot 95\,500}{340{,}4 \cdot (1{,}1 \cdot 19)/2}$	$F = \dfrac{768 \cdot 95\,500}{127 \cdot (1{,}6 \cdot 19)/2}$
F	$F = 11\ 937$ kg	$F = 21\ 048$ kg	$F = 37\ 994$ kg
z_1	$z_1 = 20$	$z_1 = 19$	$z_1 = 19$
X_1	$X_1 = +0{,}5$	$X_1 = +0{,}3$	$X_1 = +0{,}2$
q	$q = 2{,}6$	$q = 2{,}75$	$= 2{,}96$
b	14,0 cm	20,0 cm	28,0 cm
m	0,8 cm	1,1 cm	1,6 cm
e	1,37	1,365	1,37
Q (helix angle = 12°)	1,3	1,3	1,3
σ	$\sigma = \dfrac{11\,937 \cdot 2{,}6}{14 \cdot 0{,}8 \cdot 1{,}37 \cdot 1{,}3}$	$\sigma = \dfrac{21\,048 \cdot 2{,}75}{20{,}1 \cdot 1{,}1 \cdot 1{,}365 \cdot 1{,}3}$	$\sigma = \dfrac{37\,994 \cdot 2{,}96}{28 \cdot 1{,}6 \cdot 1{,}37 \cdot 1{,}3}$
	$= 1556$ kg/cm^2	$= 1482$ kg/cm^2	$= 1409$ kg/cm^2
$\bar{\sigma}_{\text{allowed}}$	$\bar{\sigma} = 2000$ kg/cm^2	$\bar{\sigma} = 2000$ kg/cm^2	$\bar{\sigma} = 2000$ kg/cm^2
$f_s = \dfrac{\sigma_{\text{all}}}{\sigma}$	$f_s = 1{,}28$	$f_s = 1{,}35$	$f_s = 1{,}42$

9.11 The Promo-Teus Conveyor Belt System

Halmij BV, situated near Gorinchem in the Netherlands, has developed a fully patented Modular Container Conveyor Belt System for the internal transport of containers on a terminal.

The system is built up of standard modules; one module for length transport and one module for transverse transport. When used in combination with special intersections the complete horizontal transport of containers over a terminal can be arranged. Because of the modular design all containers on the system can have completely different and independent routings at the same time.

The system is fully scaleable and can thus be used for either large or small operations. Each container is carried via the four corner castings by the two, specially designed narrow belts. The ruggedized belt construction allows a smooth transfer of the containers between the belts. The belt speed is variable; the maximum speed of the container is approximately 1 m/sec. Figures 9.11.1 to 9.11.3 show a test site of Promo-Teus.

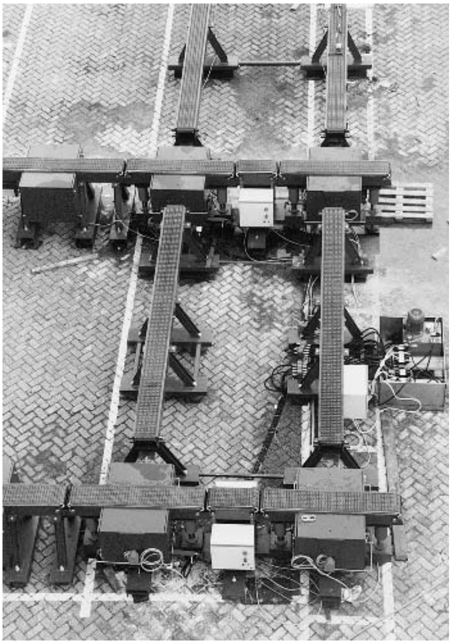


Fig. 9.11.1 Promo-Teus

Fig. 9.11.2 Container being loaded

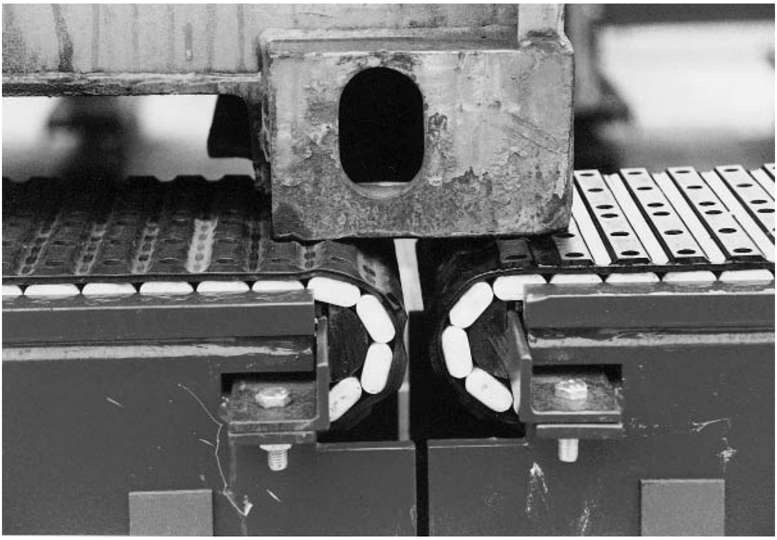


Fig. 9.11.3 The belt system of Promo-Teus

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Chapter 10

Maintenance

General

With a well made piece of equipment, maintenance becomes a major factor to keep this machinery in good condition. An organization with reliable maintenance engineers should be formed to do this important job. Discipline is needed to carry out regular inspections at the right time and with the necessary care and attention.

For rolling equipment like straddle carriers and AGVs (Automated Guided Vehicles), a well equipped workshop will be the best place to concentrate all important maintenance jobs.

Moveable platforms which can surround the taller equipment such as straddle carriers can be useful, as can moveable grease guns with long, flexible grease hoses and moveable drain containers. Special cricks can help to change heavy tyres rapidly and easily.

The extensive maintenance manuals normally give enough information about the frequency of inspection and the items which are to be checked and maintained. The same principles apply to mobile cranes which are able to move around freely. However, as they are normally too large to be worked on inside a workshop, the maintenance must be carried out *in situ* or in a predetermined maintenance position at the terminal or quay. Refuelling of the diesel engines also needs to be organized with precision.

For cranes running on rails, such as the many types of ship-unloading and loading equipment, stacking cranes, etc. the maintenance work must be carried out *in situ*. The complete systems and the automation require specialist skills. The training of a suitable team of operatives is expensive and time consuming, but absolutely necessary. Inspection and maintenance of the hydraulic equipment similarly demands specialist knowledge and a sound understanding of the systems which are in use.

Mechanical engineers must inspect the wire ropes and wire rope systems, hoist-, travel-, luff- and slew mechanisms, brakes, gearboxes, and drums. Steel structures also require checking for fatigue cracking and other faults. Bolts should be checked regularly for signs of loosening, corrosion, cracking or other damage. Greasing and lubrication are an important part of this whole process because while this essential procedure is being undertaken, the engineers can carry out visual, mechanical and other testing at the same time. Greasing and lubrication need to be thorough and not skimped on even though it is both costly and messy. It is one of the most important ways in which the useful life of equipment can be extended and the downtime through repair and breakdown reduced.

Railtracks should be inspected from time to time, especially those tracks which are laid on sleepers and ballast beds.

– Allowable deviation of the span	if span $\leq$ 15 m, $\Delta = 3$ mm if span $\geq$ 15 m, $\Delta =$ to 10 mm increasing
– Allowable deviation of one rail from the nominal straight line in the horizontal plane	max 1:1000
– Allowable deviation of one rail from the nominal straight line in the vertical plane	local 1:1000; over the full length of the track 1:5000.

Maintenance manuals

Maintenance manuals should comprise:

- Introduction
- Technical main-characteristics
- Warning about the windspeeds in which the equipment can work and should be locked against storms, etc.
- Safety demands and safety procedures
- General warnings; signals to be used
- Instructions for the use of fire-fighting equipment, etc.
- All sorts of drawings and information for instruction, layout of mechanisms, etc.
- Certificates for the wire ropes

– Instructions for the use of oil and grease
– Intervals between inspection and control of mechanical and electrical parts
– The instructions for the inspection of steel construction parts and their conservation
– The allowed tolerances of the rail-tracks
– (The electrical system is normally described in separate maintenance books.)

Maintenance periods

Regular maintenance is most important and will lengthen the lifetime of the equipment, lessen the downtime and prevent damage. Maintenance means, among others, a good greasing of all mechanisms, and also a regular control of such simple items as bolts and nuts. An extra inspection is recommended after a heavy storm.

Inspections should be 'visual' when the mechanisms are at rest, and also when they are working. It is also necessary to check the motor-, coupling-, gearbox- and brake-temperatures during working and the wear-and-tear of small items such as brake-pads and brake-linings.

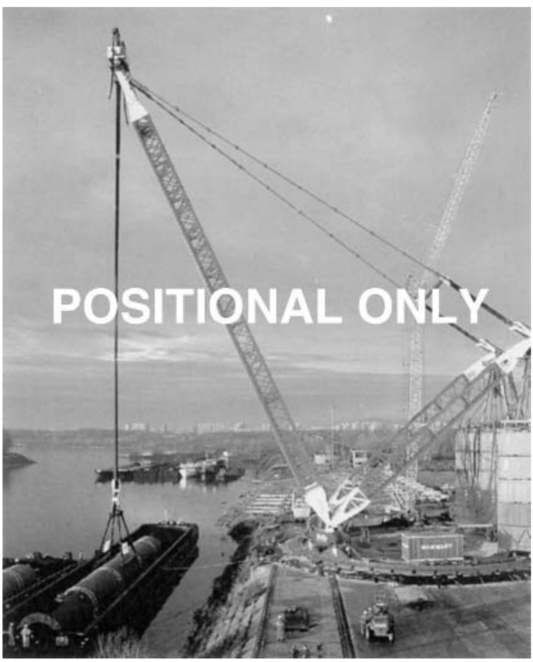

Fig. 10.1.1 2000 ton erection crane

The general lifetime of wire ropes has been mentioned in Section 2.7. In general the wire ropes of heavy duty, very frequently used cranes will have a restricted lifetime. Mechanical damage through hitting cells and hatches often occurs. Boom-hoist wire ropes have a lifetime of 5 to 8 years.

	Mechanical			*Electrical*			*Lubrication*				
Preventive maintenance	1 × per week	1 × per month	1 × per 3 months	1 × per week	1 × per month	1 × per 3 months	1 × per month	1 × per 3 months	1 × per 6 months	1 × per 12 months	When necessary change oil (c.o.)
Hoisting mechanism											
gearbox		×									x (c.o)
couplings		×						×			
brakes	×						×				
overload preventer		×			×			×			
wire ropes	×							×	×		
wire rope drums		×						×	×		
guide rollers	×						×				
wire rope sheaves	×								×		
motors & electrical parts				×					×		
Trim; list; skew mechanism	×			×							×
Anti-pendulum system	×			×							
Trolley travelling mechanism(s)											
gearbox		×									× (c.o.)
couplings		×						×			
brakes	×						×				
wheels; rails		×							×		
wire ropes	×							×	×		
guide rollers	×						×				
wire rope drums		×						×	×		
wire rope sheaves	×								×		
motor(s) and electrical parts				×					×		
Boom hoist mechanism											
gearbox		×									× (c.o.)
couplings		×						×			
brakes	×							×			
emergency brake	×							×			
wire ropes		×							×		
wire rope equaliser		×									

Preventive maintenance	*Mechanical*			*Electrical*			*Lubrication*				
	1 × per week	1 × per month	1 × per 3 months	1 × per week	1 × per month	1 × per 3 months	1 × per month	1 × per 3 months	1 × per 6 months	1 × per 12 months	When necessary change oil (c.o.)
Boom hoist mechanism (cont.)											
wire rope drum		×							×		
wire rope sheaves		×							×		
boom latch		×						×			
motor & electrical parts				×					×		
Crane travel mechanism											
gearboxes		×									x (c.o.)
open gearings		×					×				
couplings		×						×			
brakes		×						×			
wheels		×							×		
rails			×								
buffers			×						×		
anchoring device			×								
motors & electrical parts			×						×		
Change oil in all thrustors; oil pumps; emergency brakes										×	
Elevator											
mech. & electr. parts	×			×							
pin & rack							×				
wire ropes								×			
Fire fighting equipment			×								
Crane drivers & other cabins											
mech. and electr. parts					×						
suspension of the cabin			×								
M.V. cable drum and cable											
mech. & electr. parts	×			×							
Steel structures											
visual inspection		×									
bolts & nuts (visual)			×								
fatique (visual)			×								
cracks (visual)			×								
painting (visual)			×								

	Mechanical			*Electrical*			*Lubrication*				
Preventive maintenance	1 × per week	1 × per month	1 × per 3 months	1 × per week	1 × per month	1 × per 3 months	1 × per month	1 × per 3 months	1 × per 6 months	1 × per 12 months	When necessary change oil (c.o.)
Machinery & Electrical Installation houses											
Aircons, ventilation and heating		×			×			×			
maintenance crane			×			×				×	
compressor			×								
electrical cubicles				×							
transformers, etc.					×						
Spreaders	×			×			×				
Grabs; motor grabs	×			(×)			×				
Boom hinge points		×					×				
Hinges in boom ties; sheave blocks, etc.		×					×				
Emergency drives			×			×					× (c.o)
Festoon system		×			×						

The first time that the oil in gearboxes has to be changed is after some 500 working hours. Check then carefully whether mineral oil or synthetic oil has been used in the gearboxes and follow the guidelines which the oil-companies have given.

Slewing- and luffing-mechanisms

The same sequence of maintenance can be followed as for the trolley-travelling mechanisms.

The rollers of double-row ball bearings in slewing mechanisms should be greased after each 50 working hours. It is advised to use automatically and continuously working grease pumps.

Fig. 10.2.1. Multipurpose Mobile Feeder Server

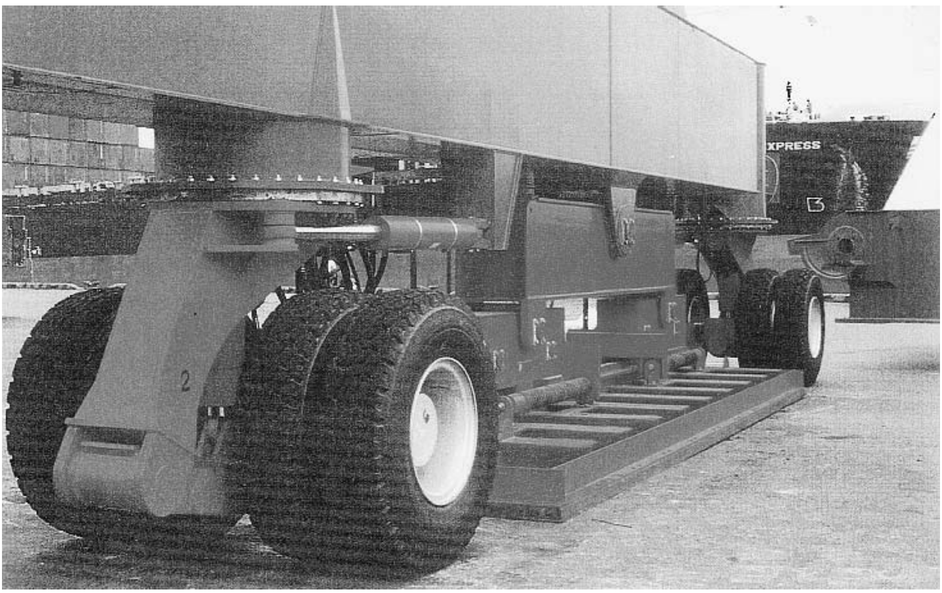

Fig. 10.2.2 Multipurpoe Mobile Feeder server; rubber tyred crane drive units and load support segment

Fig. 10.2.3 Kalmar Container Crane with 70 m span

Artwork Sources

Chapter 1

Fig. 1.1.1	Creative problem solving	R. A. Kujawski and L. L. Smith, *Chemical Engineering*, 18 July 1988
Fig. 1.1.2	Wooden crane	Visserijmuseum, Vlaardingen Photo A. Wapenaar
Fig. 1.1.3	Driving the hoist of Fig. 1.1.2	Visserijmuseum, Vlaardingen Photo A. Wapenaar
Fig. 1.1.4	The development of slewing level luffing cranes from 1856–1956	Maritiem Museum Rotterdam
Fig. 1.1.5	The development of floating cranes 1905–1936	Maritiem Museum Rotterdam
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Fig. 1.1.7	Lemniscate grabbing crane	Nelcon
Fig. 1.2.1	2500 t offshore crane	Gusto/Kone
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Fig. 1.3.7	Ringer crane	Mammoet. van Seumeren
Fig. 1.3.8	Floating derrick	Hong Kong
Fig. 1.3.9	Two offshore cranes; each for 4000 t (by courtesy of Heerema/IHC Gusto Eng. BV)	Heerema/Gusto Eng.
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Fig. 1.3.11	Container mover	Tracon, Rotterdam
Fig. 1.3.12	Excavator	Caterpillar
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Fig. 1.4.1	Break-down of ship and gang times (by kind permission of Dr A. Ashar)	Dr A. Ashar, Lousiana State University, U.S.A.
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Fig. 1.4.3	Production scheme	Author

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Fig. 6.12.3	Crane with laser scanners	Lase GmbH; ZPMC
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Fig. 6.14.2	Overview of the system	Orlaco

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Fig. 7.2.2	Welded pipe constructions	Nelcon
Fig. 7.2.3	Reduction of the sensitivity to lamellar tearing	Author
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