POLITECNICO DI TORINO

Department of Environment, Land and Infrastructure Engineering

Master of Science in Petroleum Engineering

Design of Cranes for Industrial Lifting Operations.

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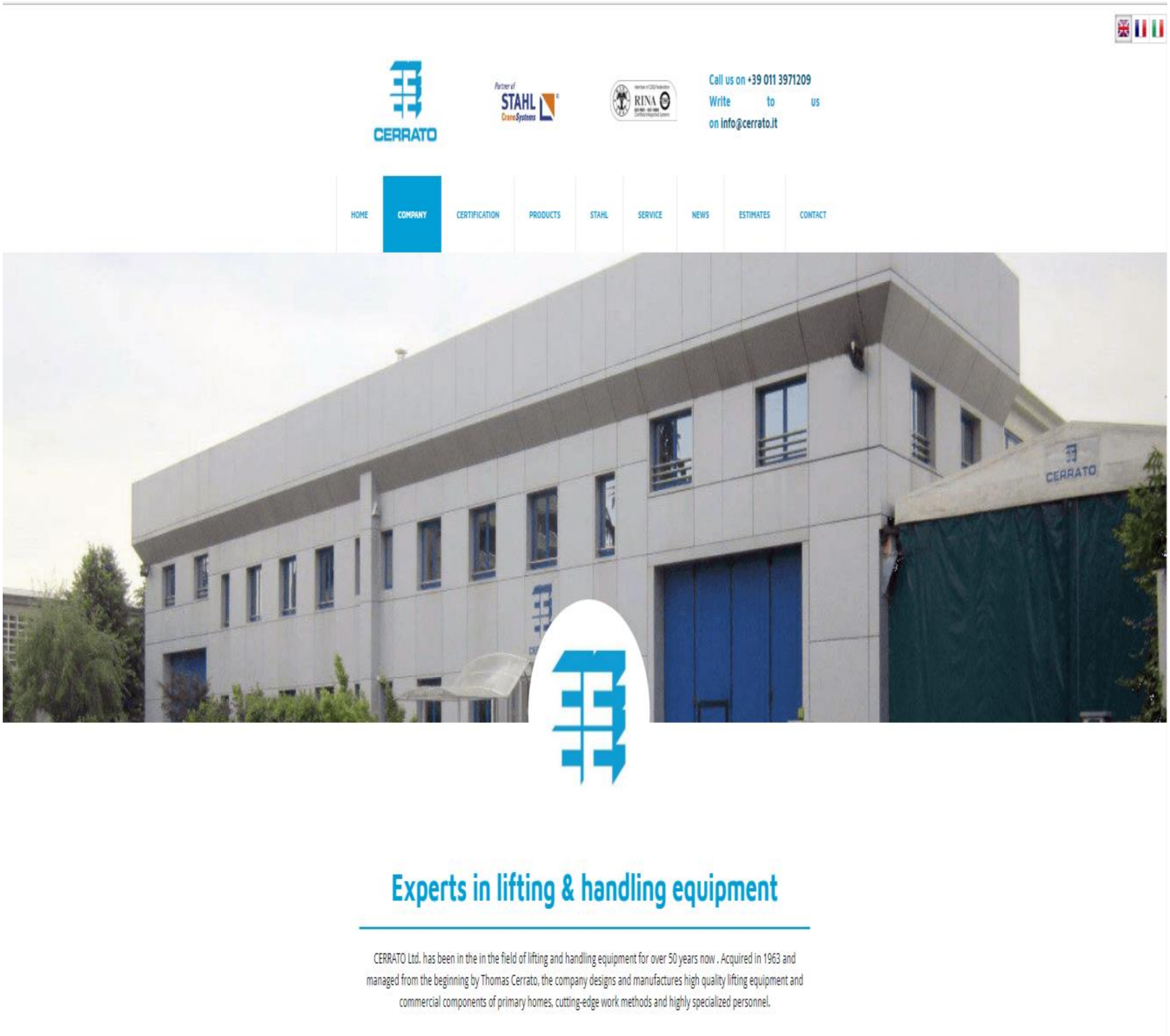

Figure 1: CERRATO s.r.l Company Headquarter BEINASCO

Figure 2: CERRATO Company Official Partners

Figure 3: Crane Components

ABSTRACT

CERRATO Company is located in BEINASCO. Headquarter is shown in Figure 1.[1]

Partners of the Company are mentioned in Figure 2. [2]

In the thesis, the Design of Crane in CERRATO s.r.l Company is explained using a Design program developed in the Company's Technical Office.

The Design is made for the Crane Components shown in Figure 3. [3] In order to present a Clear Design all components of Crane are described separately.

The Crane is needed for industrial applications intended to lift 33 tons. The case is applied in CERRATO COMPANY. [4]

The objective is the application of the program to define the specifications needed for all components composing Crane. A series of calculations will be conducted to find all Dimensions needed to have a complete Crane system ready to be installed for Clients.

In the case reported below the number of drums is 1. Number of pulls from drum sides is 2. Number of crane rope falls is 8. These three parameters are shown clearly in Figure 4. [5]

The Results needed are the right specifications within the correct safety margins chosen by engineers.

Spreadsheet is divided into different sections for clearer explanation. In each section every independent parameter is assigned a value.

Either these values are imported from Company Database or calculated using formulas defined in later sections. Company's Database includes Technical Data, Commercial Data, Catalogs, History Data and Normative applied.

Verifications are made to make sure all conditions are within limits set, if not disasters can occur.

The sequence of steps to be followed is consistent and verified. For easier application, every element of hoist system will have its own design spreadsheet.

Afterward, all elements can be combined to form the system wanted. Client gives the input depending on his own needs and application of the crane (example given Power plant, food industry …).

The basic step for the Design procedure startup is the knowledge of application type. Each type of application has its maximum lifting weight and work duration. Those data are very important for the initiation of the design.

Most of hoist systems are alimented by electrical current from electric motors. For a specific conditions of crane construction correspond a specific gear and motor that must be employed properly. If any component of the system fails, else the whole system will not work.

The system shown is initially presented in a general way. In further sections it is will be described in details.

A fatigue analysis is conducted at the end of the study to verify that components dimensions are correct. All the standards used were based on technical team experience.

Figure 4 represents a simulation of the case studied aiming to explicit the Crane notion for Readers.

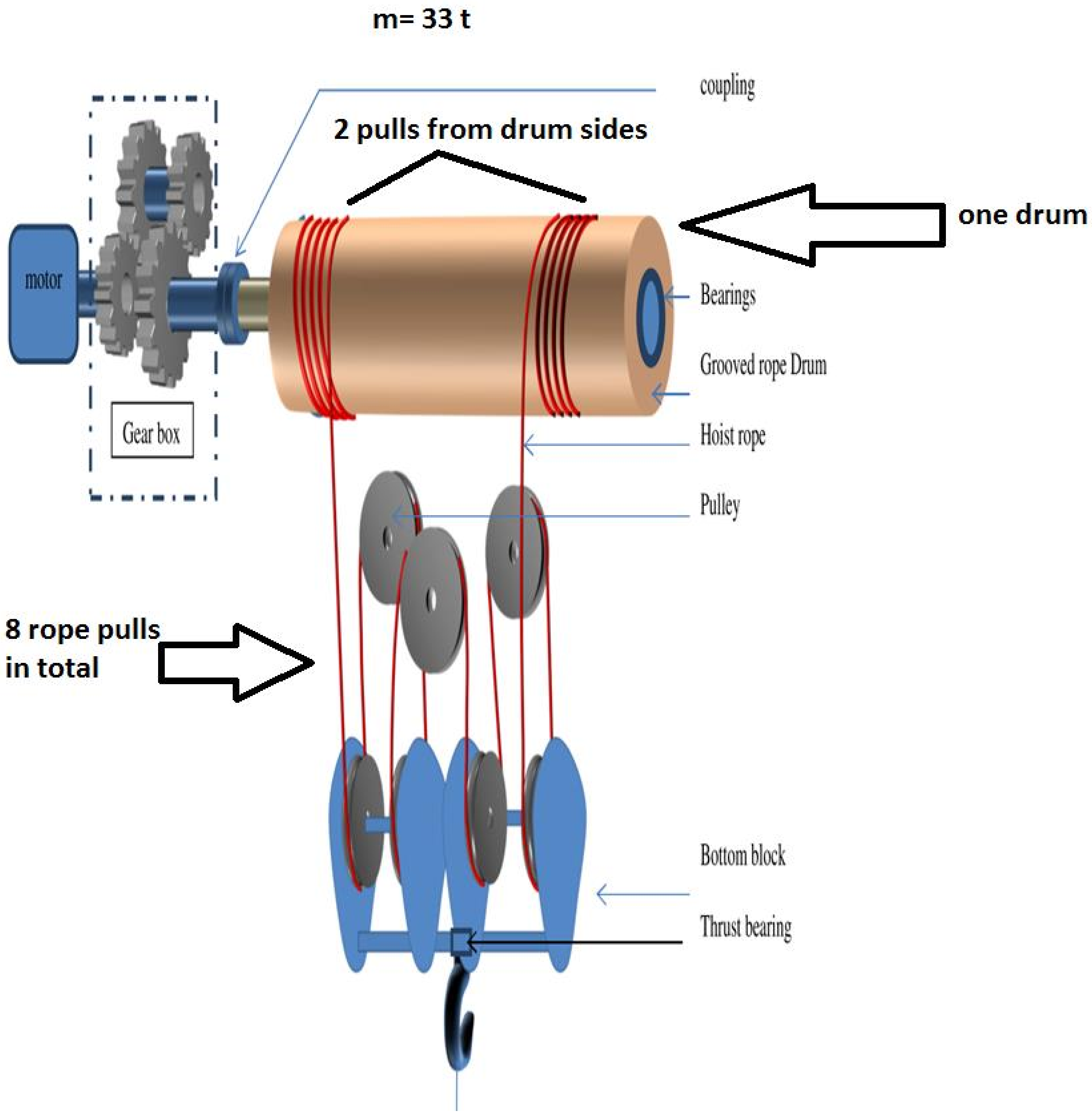

Figure 4: Scheme of Case Studied

ACKNOWLEDGMENTS

All source of information were exploited using legal access in the Company’s Technical office including books, Normative, and Catalogs. [6]

Internet sources were also used in addition to the information given by experienced engineers working in technical office.

All References are cited including all images inserted.

The Thesis is dedicated to the R&D Center for the CERRATO s.r.l Company.

The work was made in the CERRATO s.r.l Company Technical office with collaboration of the technical team made of technicians and engineers.

Engineer Leonardo UCCIARDELLO and Engineer Carlo MARCHIO supervised the work in CERRATO Company.

Professor SURACE supervised the work in Politecnico di Torino.

All data comply with CERRATO s.r.l Company’s Rules and Restrictions. [7]

CERRATO’s domain of application exclude Oil Rigs Cranes for which special designs are made.

The Thesis topic was selected after discussion with Engineers Leonardo UCCIARDELLO and Carlo MARCHIO.

The Requirements were made according to Company’s Syllabus independently from the Degree Specialization.

INTRODUCTION

A Crane is a mechanical device used to lift mass from ground upward and vice versa.

“Need is a demand and demand needs a supplier”.

This was the reason of Crane invention & Crane constructing Companies foundation. It was the result of industrial need called lifting.

Lifting is necessary to accomplish an infinite number of tasks like commissioning and decommissioning of heavy mechanical machines, which cannot be lifted manually by Human beings.

Today, almost every industry possess a crane system to carry heavy loads and to move them.

However, being installed in a very wide range of industries, cranes are used in a continuous manner in some of them whilst in the other they are rarely needed.

For this reason, a criterion is defined in Crane design representing the frequency of Crane functioning in order to differentiate an application from another.

CERRATO Company manufacture Hoist systems for clients depending on their need and their abilities.

The company askes the client for their application for which the hoist system is needed so the design procedure of crane system starts.

Traditional used methods defines the type of crane needed for every application based on the experience of engineers and technicians present in the company.

The main CAD programs used in CERRATO for the crane design are INVENTOR and AUTOCAD 3D.

CHAPTER 1: GENERAL SYSTEM DESCRIPTION

Figure 1.1: Crane 3d Simulation CERRATO S.R.L

Figure 1.2: Crane Construction at CERRATO

1.1 GENERAL DEFINITIONS

1.1.1 RAILS CONSTRUCTION

In Figure 1.1, a complete 3D Crane System is shown after being designed. [8]

Two rails are constructed under the sealing of the factory on which the Hoist system will displace. An example is shown in Figure 1.2. [9]

The most common type of section used is the I beam. In construction, the bending moment is more relevant than shear forces because it is the principal reason for beam collapse.

Therefore, the design is made mainly considering bending moment resistance. The I beam provides a Good Bending moment resistance since due to its form the two flanges resists more than 80 % of the bending moment.

The material concentration is close to the most stressed area and thus represents higher resistance compared to other shapes such as rectangular sections. [10]

After constructing the two supporting rails, the hoist system beam can be installed perpendicularly.

The hoist system motion is restricted to the horizontal direction in the 2D plane parallel to the floor plane and the vertical line perpendicular to this plane.

The three directions in which the system motion is allowed are show in Figure 1.3. [11]

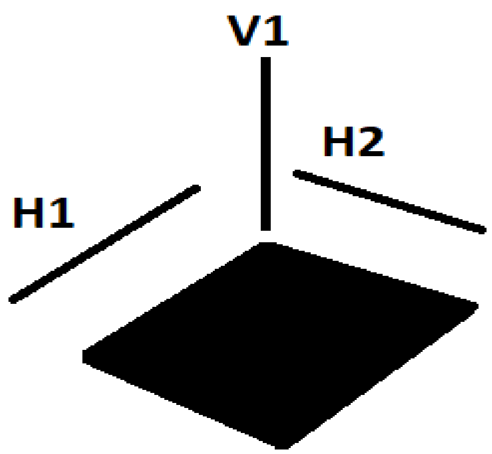

Figure 1.3: Allowed Motion Directions

As shown in Figure 1.3, in the horizontal plane, the Crane System can move in two perpendicular directions H1 & H2 one at a time and the course in the two directions is defined by application needed (span to be covered). Concerning the vertical line motion, the trolley is only moveable along V1 and has a defined course.

1.2 SYSTEM TRANSLATION

1.2.1 TRANSLATION POWER

The main challenge of Crane system displacement is to overcome attrition forces, for which are subjected crane components. Because of heavy weight lifted, load stability assurance, heavy bearing, and Gravity forces must be well controlled. The translation power in system is the power needed to move the whole system in translation along rails.

It mainly depends on forces described before acting on system, related to it's mass (cart mass, bridge mass, load). The relation is represented in the equation below:

$$\sum \boldsymbol{F} = \boldsymbol{m} \times \boldsymbol{a}$$

One of the most important parameters to specify is the Hoist system vertical velocity. Each type of application has its own translational velocity.

It is needed to determine parameters such as rope length, drum diameter, pitch, motor & reducer choice, torque to be applied etc. [12]

The translation of the system can be represented by simple Cinematic equations.

1.2.2 DISPLACEMENT DETERMINATION

Supposing x is the abscissa along the whole race, v is the velocity of the translating system, a is the acceleration.

Acceleration will be constant a=cst.

Thus, the system will be uniformly accelerated (or decelerated depending on the direction of movement) ,

Knowing that the acceleration is the derivative of velocity, the integration can be used to find v using acceleration. The velocity will be a linear function of time:

$$\boldsymbol{V} = \boldsymbol{V0} + \mathbf{a} \times \mathbf{t}$$

V0 is velocity at time $t=t_0$; t is variable time; a is acceleration.

Knowing that the velocity is the derivative of abscissa, the integration can be used to find x using velocity.

The abscissa will be a second-degree function of time:

$$\boldsymbol{X} = \mathbf{X0} + \boldsymbol{V0} \times \boldsymbol{t} + \mathbf{a}/\mathbf{2} \times \mathbf{t}^{\mathbf{2}}$$

X0 is abscissa at t=t0; V0 is velocity at time $t=t_0$; t is variable time; a is acceleration.

The angular velocity is related to the translation velocity by the following formula:

$$\boldsymbol{V} = \boldsymbol{\omega} \times \boldsymbol{R}$$

R is the radius of the wheels (identical for all four wheels); ω is angular velocity.

When Wheels are larger the velocity increases for the same angular velocity.

Translation Power is shown in the formula below **:**

$$\boldsymbol{P} = \boldsymbol{F} \times \boldsymbol{V} = \boldsymbol{F} \times \boldsymbol{R} \times \boldsymbol{\omega}$$

The previous formula is used in the motor case to calculate motor power considering the angular velocity and the torque applied on the motor shaft.

In this way, the rotational power of the motor is converted to translational power allowing the system to move in translation.

The input power must exceed the required one, or else the rotating shaft will not work.

In our spreadsheet, the reducer choice will be based on Reducer Couple, and service factor needed.

The service factor is mainly preferred to be around 1.3.

After satisfying the required criteria, the motor and the reducer must be implemented together in the whole system.

1.2.3 COMPONENTS IMPLEMENTATION IN WINCH SYSTEM

The drum is not present in all Hoist types i.e. Tower Crane systems. It is commonly used in Winch systems with other components indicated in Table 3. [13]

Mainly it is employed where the load need to be lifted vertically for the following advantages:

-Distortion of Rope is minimal.

-Load is lifted vertically while being horizontal which represents the most stable case of lifting.

-Inclination is minimal, this is critical parameter that can lead to load instability and fall.

-Heavier loads can be bore because of the advantages presented above.

Drum is always installed in the same position in Winch systems. A part is connected to the rotating shaft, the other to Bearings.

The shaft of the drum is studied and chosen carefully for the specific application type. Generally, the material type is unique and will be defined in other sections.

Supports and pins are implemented on shaft to block components and assure static stability.

The ropes are directly related to the drum or the tackle. It transmits load via hook and pulleys.

Ropes are flexible for manipulation but presents a higher risk compared to chains.

A gear coupling is selected to connect reducer and rotating shaft. It is the joint between the gear motor and the drum.

Bearing are also designed properly for the installation of the drum inside the crane system.

Note that the Company buy the tackle and doesn't manufacture it. [14]

Usually tackles are used for moderate loads while Winch are used for severe cases.

To determine all the above-mentioned elements, data shown in Table 1.1[15] are entered to the Design program explained in latter sections. The Results of the Design will be the data listed in Table 1.2.[16]

Combination of Elements of the Crane in CERRATO Company is shown in Figure 1.4[17] and Figure 1.5[18].

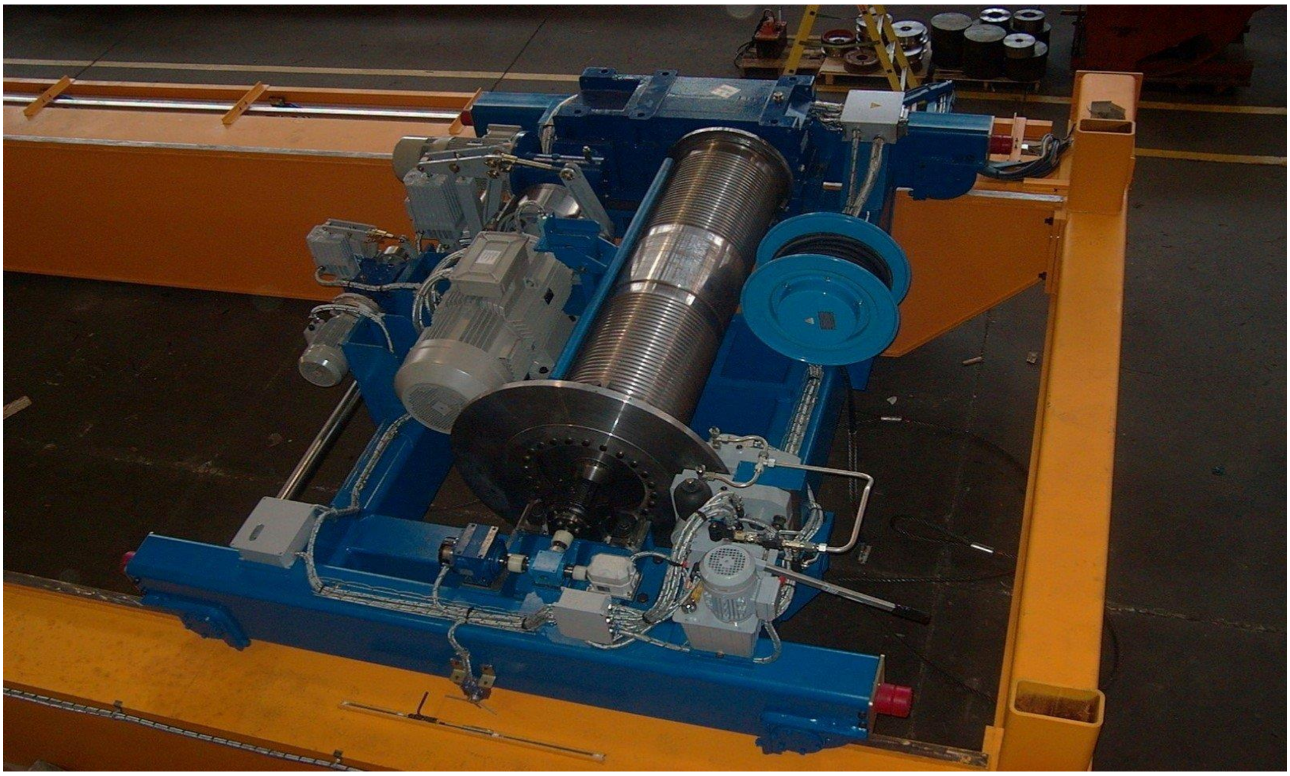

Figure 1.4: System installation CERRATO Industry BEINASCO

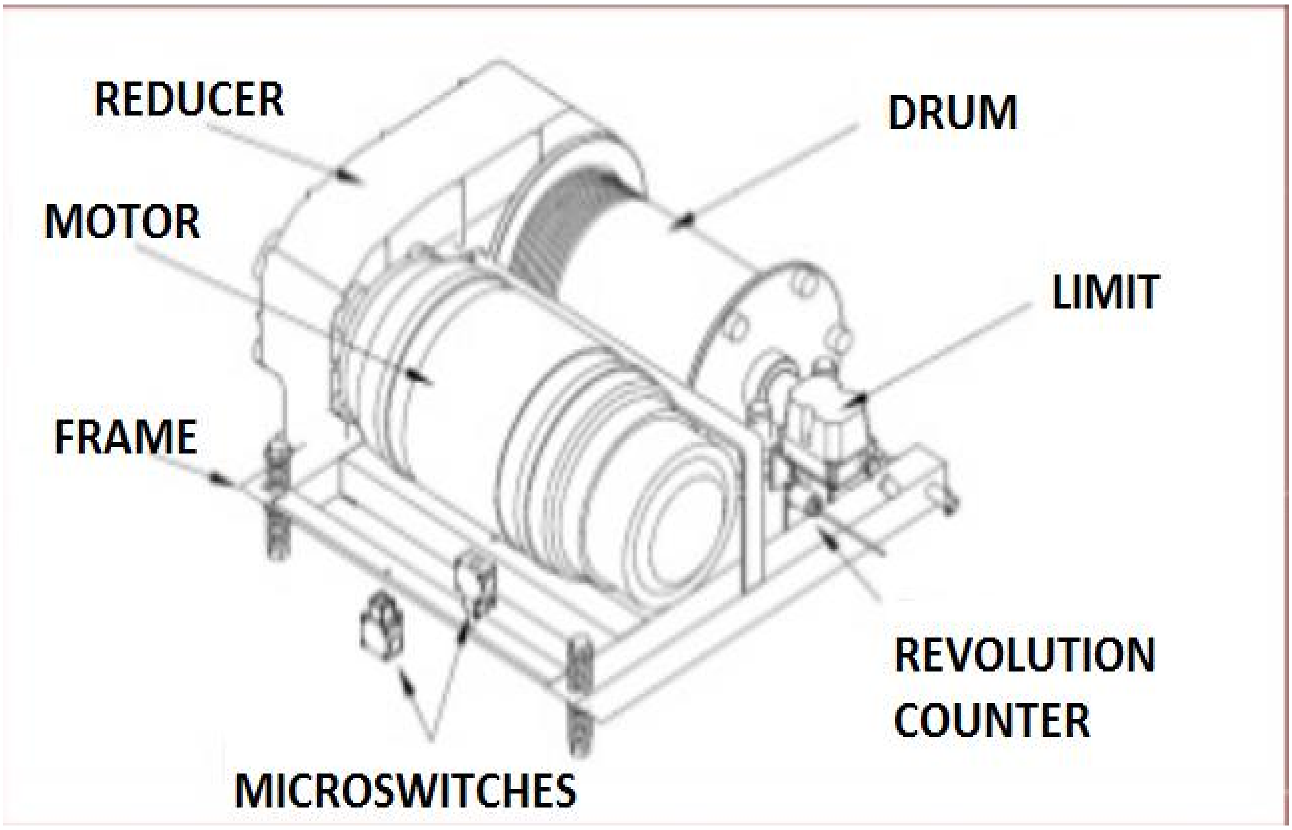

Figure 1.5: Winch components

1.3 INPUT and OUTPUT DATA

Table 1.1: SYSTEM INPUT DATA

M1: Bridge mass Kg
M2: Cart mass Kg
M3: loading mass Kg
Sc : beam width m
D : wheels diameter m
M : Class of mechanism used
Va : translation velocity m/s
Ds : shaft diameter m
X : number of motors
Nrm/Nrt: ratio of driving wheels over total number of wheels
C/B :Type of rail on which are wheels mounted
B/R : Type of wheel connection
Y/N : External reduction of reducer
Z1 : number of pinion teeth
Z2 : number of crown teeth
Motor Efficiency
Motor Type
Mf : Braking torque N*m
R.R.e: Effective Reduction ratio

Table 1.2: SYSTEM OUTPUT DATA

Rope material type and diameter
Rope stress resistance
Drum material type and diameter
Pulleys diameter
Shaft diameter
Shaft number of cycles
Motor and reducer type
Motor and reducer Power
Bearings type and dimensions
Bearings number of cycles
Couplings type
Couplings size

Table 1.3: WINCH COMPONENTS

Ropes
Drum
Rotating Shaft
Bearings
Frame & Supports
Reducer
Electric Motor control unit
Gear couplings
Motor Brake

1.4 COMPONENTS DESCRIPTION

1.4.1 ROPE

The rope is responsible for the connection of the loaded system to the upward bridge system. Its design is the fundamental to maintain stability of system and to design correctly the other Hoist system components. It is described lately in details.

1.4.2 DRUM

Wrap-around drum made of a high strength and turned steel tube to obtain a double helicoidally thread.

Drum Material type is chosen to withstand all stresses subjected and at the same time to undergo right welding operations in parts where section variations exists.

The drum is controlled by the slow shaft of the lifting gear through a broached coupling of maximum reliability.

1.4.3 SHAFT

The shaft is the unit responsible for transmitting torque from gear motor to the drum. In addition, it is the unit holding the drum and other components. It is connected to the gearbox via gear couplings.

1.4.4 MOTOR

The motoring system is composed of a three phase asynchronous motor (no need to have direct connections to produce electricity, the electric production is made by rotation of rotor in the stator frame).

The rotor used has a conical shape and the brake of the motor is of electromagnetic type. The motor is flanged directly to the gearbox.

The main unit and the corresponding group have easy access; they can be dismantled fast without the intervention on other parts of the machine.

Reducer: Reducer is a special type of gearbox in which the angular speed is always reduced.

1.4.5 BEARINGS

The bearing principal function is the friction reduction between two rotating systems at the interface. The Bearings assure the synchronal rotation of the second side of the shaft with the first side connected to the gearbox.

1.4.6 GEAR COUPLINGS

Gear Couplings are generally used to connect two rotating shafts, which have to transmit a torque in a torsional rigid way.

Couplings requirements must be in accordance with the following list of parameters: angular, parallel, axial, combined misalignments, maximum speed rotation, and transmitted torques.

Cerrato Company uses traditional catalogs made by MAINA Company. [19]

1.5 APPLICATION OF CRANE SYSTEM IN OIL OFFSHORE RIGS

Figure 1.6: Offshore Cranes

1.5.1 AREA OF APPLICATION

Heavy lift

Generally Offshore crane systems like the one shown in Figure 1.6 are designed to bear very heavy loads compared to the conventional industries. Crane systems are responsible for assembly, Commissioning, decommissioning of Rigs and heavy towers. Plus, the implementation of huge storage elements such as gas storage units and liquid oil. Lifting capacity of offshore cranes can reach 2000 tons.

Hence a lot of challenges are encountered in oil Industry in which safety and precision are the Priorities.

High altitudes

Altitude in offshore rigs is much higher than conventional industries because of huge units implemented. Capacity of units exceeds the capacity of onshore units for Economic Reasons.

Altitudes of Offshore cranes can exceed 100 m.

Deep sea

Special equipped Crane system is used in deep seawater rigs to setup Oil and Water wells. For this reason, several types and designs of Cranes are present as shown in Figure 1.7 to be flexible with the proper application required. [20]

In this Thesis Design does not include Offshore Crane Systems because it has a different scope from CERRATO COMPANY, which is limited to conventional industrial applications. [21]

Figure 1.7: Offshore Crane types

CHAPTER 2: CRANE COMPONENTS DESIGN

2.1 ROPE DESIGN

Figure 2.1: Steel Rope 3D view

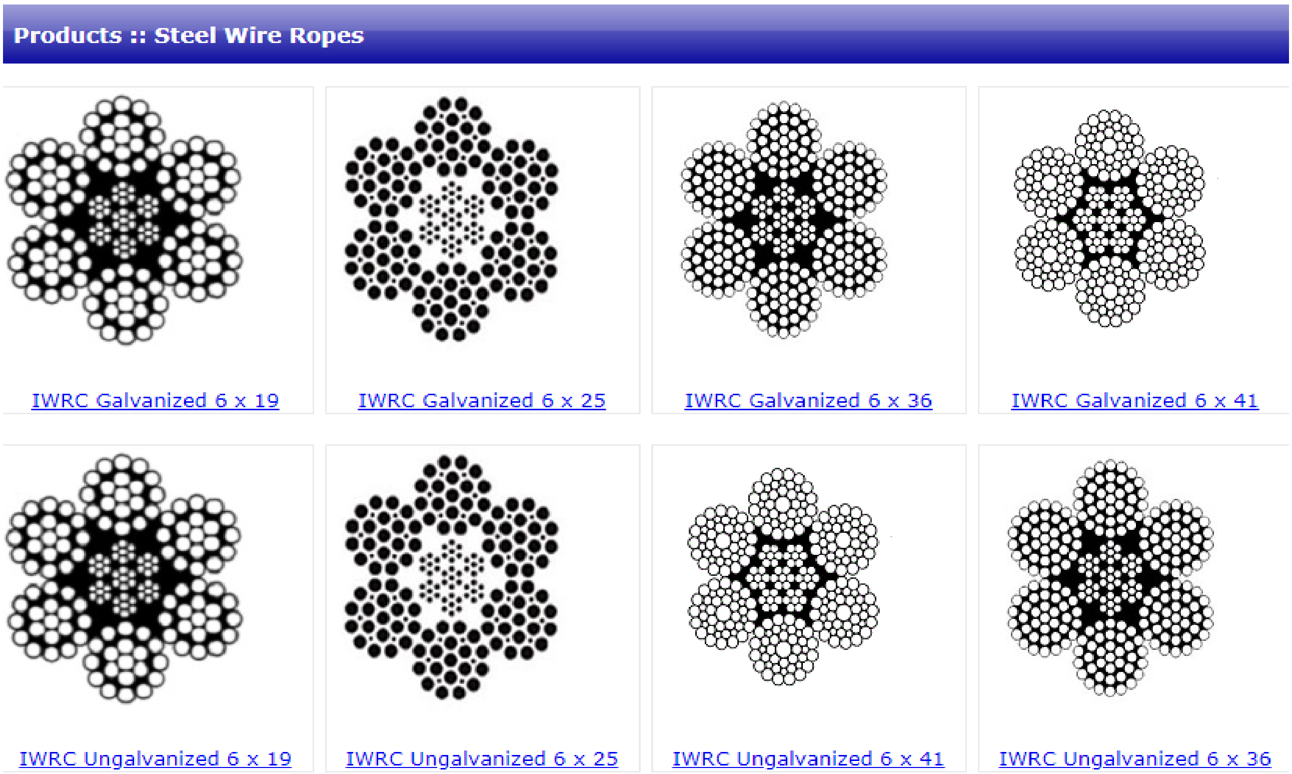

Figure 2.2: Different Rope types

2.1.1 INTRODUCTION

The Ropes is the fundamental mechanical component of a crane device used to connect lift to winch.

The main type of material used for Crane ropes are steel. The main application for ropes is in civil engineering (construction ….) and mechanical engineering (Hoist systems…)

The dimensioning of the rope means determination of its appropriate diameter. It is obtained after calculation of the maximum tension for which the rope will be subjected.

All rope calculations are shown later in details.

An illustration of the Steel Rope in 3D is shown in Figure 2.1. [22] The different rope types are present in Figure 2.2 .[23]

The following parameters are checked before choosing the rope:

Breaking strength, minimum breaking strength, and safety factor. [24]

2.1.2 STANDARDS SELECTION

The standards used in Ropes application will be standards of Teci's company. It is the leader in Italian market of steel wire ropes and accessories to lift.

It produces ropes of different types and dimensions. End fittings for the ropes are also made.

Teci has a wide range of ropes and accessories depending on the customer's need.

Teci has a Test laboratory to verify failure and tolerance of ropes given a certificate of quality inside Italian and international standards law. Teci's product are mainly used in construction field.

All the ropes have warranties and certificates compatible with Italian law .They can be tested by public and private institutes. [25]

2.1.3 ROPE PARTS

a-Wire

The main component of the rope it is composed of a wire rod galvanized in a hot dip galvanizing process.

b-Strand

After placing the central wire it is covered helically with wires to form the strand of the wire.

c-Rope

The rope will be the union of the two previous components and will have the following advantages:

In case the rope is cut the strands are not affected so there's no need to weld them.

The lifetime of ropes is increased.

The wires will not displace even though if they break.

2.1.4 MAIN PROPERTIES OF WIRE ROPES

Diameter

It has to be distinguished between the different type of diameters used in ropes lexicon. When the word diameter is used alone , it is meant the normal diameter of the rope.

Lay direction

According to ISO 2408 lay direction means the direction of the outside wires compared to the strands direction and the strand compared to rope.

Breaking force

Breaking force is divided into different types depending on the breakage situation.

Minimum breaking force

It is present in catalogue for each specific rope type. Used to prevent accidents breakage to set the load capacity needed.

a-Actual Breaking force

It is found by applying a breaking test. It is superior to the minimum breaking force.

b-Aggregate breaking force

It is the sum of single wires break force before starting strand (it is theoretical). It is superior to Actual breaking force and minimum breaking force. It cannot be used to calculate load potential. [26]

2.1.5 PROCESSES APPLIED FOR ROPES

2.1.5.1 COMPACTION OF STRANDS

Why the compaction of strands is applied?

The answer is reported in the following:

Metallic area increase.

Increase of contact surface between wires.

More regular, smoother and less permeable surface.

Tension distribution on wire is more equal.

Dimensional stability improve against side forces.

Improve elasticity modulus.

Compaction can also increase ductility of rope and it’s fatigue resistance.

2.5.1.2 COATING CORE WITH PLASTIC

It is consisting of a plastic sheath coating the metal made rope core. The inclusion of plastic aims to reduce sliding possibility of the components and geometric variations.

2.5.1.3 PRE-STRETCHING OF ROPES

Pre-stretching treatment is used to reduce the permanent elongation.

Pre-stretching has two types :

Static Pre-stretching : applied at half breaking force magnitude. Application of a series of load cycles (loading and unloading cycles)

Dynamic Pre-stretching : applied during steel wire close process. The applied force is 1/3 breaking force magnitude. This type presents an actual break of the rope.

Main advantages

Reduce internal corrosion : When plastic is inserted the penetration of polluting agents is more difficult and thus permeability of wire is reduced causing reduction of internal corrosion.

Building a mechanical joint between the different components.

Prevent wear by reducing free space inside exterior strands.[27]

2.1.6 ROPE SPECIFICATIONS

Elongation under load conditions :

$$\Delta \boldsymbol{l} = (\boldsymbol{L} \times \boldsymbol{F})/(\boldsymbol{E} \times \boldsymbol{S})$$

where △l is elongation, L is rope length, F is force applied on rope , E elasticity modulus and S rope metal section. This formula is derived from Hook's law of elasticity.

Rope connection with Sheaves and winches :

For each type of rope used corresponds a right type of groove chosen. The compatibility between ropes ad grooves on which lies the rope is necessary to maximize rope service life.

Therefore, it starts with a correct groove dimensioning and a correct bending ratio representing the ratio between the rope and the winch diameter.

The standards used specify the minimum bending ratios.

According to FEM standards the bending ration gives the best service life expectation.

Below is presented some conditions to be satisfied by the rope and the corresponding elements.

Case of Operating winches and Sheaves :

$$\frac{\boldsymbol{D}}{\boldsymbol{d}} > \mathbf{25}; \boldsymbol{D}/\emptyset > \mathbf{300}$$

Case of Return Sheaves :

$$\frac{\boldsymbol{D}}{\boldsymbol{d}} > \mathbf{20}; \boldsymbol{D}/\emptyset > \mathbf{250}$$

Case of Lifts and Elevators :

$$\frac{\boldsymbol{D}}{\boldsymbol{d}} > \mathbf{40}; \boldsymbol{D}/\emptyset > \mathbf{500}$$

Rope Pressure on Sheaves and Winches

The specific pressure exerted by rope on groove is directly related to force applied to the rope , it's diameter and sheaves diameter.

It is calculated using the equation below :

$$\boldsymbol{P} = (\boldsymbol{T1} + \boldsymbol{T2})/(\boldsymbol{D} \times \boldsymbol{d})$$

Where P is specific pressure ,

T represents rope stress , d rope diameter and D groove diameter.

In Table 4 the Specific Pressure of different rope classes/types made of different materials is reported.[28]

Table 2.1: Specific pressure of different Rope types

Material	Cast iron G20	Steel Fe 430	Steel C45	Steel 39NiCrMo3
Rope type	Max specific pressure daN/cm2			
114 wire class	35	60	90	170
222 wire class reg.	40	75	105	210
222 wire class lang	45	82	110	220
S12-A4l-A6	50	85	120	230
S10-S11-AR-ALC	58	100	145	280

Fleet angle

When the rope is wounded down to the sheave, it is subjected to a torque on axis before arriving to the bottom. This effect is maximized when the distance between rope and sheave is short. Thus the fleet angle must not surpass 2° to avoid distortion of the rope.

Rope related variables

Composition material of rope

External diameter

Lay pitch

Lubrication type

Variables related to operating machine

Machine Material

Sheaves and winch dimensions

Bearings/bushings condition

Machine Kinematic motion

Variables related to work conditions

Work load to be applied

Dusty environment (dust concentration etc.…)

Work Temperature

Maintenance frequency

Rope rate of change

Shocks/tears gravity

Experience of previous applications and laboratories shows that the main parameters affecting rope service life are work loads and D/d ratio that must excess 16 always (generally 20 or more).[29]

Main interesting parameters in this study

Diameter of rope

Resistance to minimum breakage

Rope Filling coefficient

Minimum / calculated break stress

Combination No. of drums / No. of shots from each drum / No. of hook shots

Number of principle drums

Drum raw tube external diameter

Drum raw tube thickness

Distance between the ropes to the center (0 for 1 shot from the drum)

Rope medium distance to the center

Shaft length up to the bearing center

Distance between the drum axis and the block pulley

Distance between drum axis and pulley fixed transmission.

2.1.7 ROPE SELECTION

The ropes are subjected to ISO 2408 recommendations. The method used will be compatible with ISO 4309 “Cable surveillance “.[14]

The Rope choice will be made taking into consideration the class of mechanism.

It is advised to choose the class of mechanism superior to the one chosen for lifting. For dangerous loads M5 class is the minimal class that can be chosen.[30]

Diameter Determination

Two rope selection methods are available depending on constructor choice: [31]

A-Minimal Class coefficient for metallic cables Zp

B-Selection factor method

Common parameter for the two Methods

Determination of maximal traction effort S in lifting rope:

It is obtained considering the following hypothesis :

Maximal nominal load of hoist device,

Mitten and accessories' weight added to the load to increase the tension exerted on cable

Hauling efficiency,

Acceleration forces if they exceed 10 % of vertical load

Rope tilt at the limit of the crane if angle between rope and lifting axis is higher than 22° .

A-Minimal Class coefficient for metallic cables (Zp method)

Definition

Zp is the ratio between the minimal breaking force F0 and maximal traction effort for the cable

$$Zp = F0/S$$

The chosen cable must have a class coefficient at least equal to the minimal value Zp.

Zp can be considered as a safety factor to be precise for the selected rope.[32]

Table 5 represents the variation of Zp as a function of Class of mechanism.

Table 2.2: Zp as function of Classe of Mechanism

Class of mechanism	Minimal value of Zp
M1	3.15
M2	3.35
M3	3.55
M4	4
M5	4.5
M6	5.6
M7	7.1
M8	9

B-Selection factor method

Introduction of Method Parameters

S: maximum traction for which rope will be subjected

d: minimal rope diameter

f: filling factor of metal section

k: stranding loss coefficient

R0: minimum tensile stress of metallic wire

K': empirical coefficient for tensile stress

$$\boldsymbol{K' = (\pi \times f \times k)/4}$$

C determination :

For a chosen composition, steel, Class and minimal resistance corresponds a coefficient C defined as :

$$\boldsymbol{\left(C = \sqrt{Zp}/(K' \times r0)\right)}$$

The rope diameter must meet the following condition :

$$\boldsymbol{d \geq C \times \sqrt{S}}$$

K' can be extracted from the ISO 2408 recommendation.

Minimum Winding diameter

It is determined using the following relation :

$$\boldsymbol{D \geq H \times d}$$

D: winding diameter around pulleys and drum

H: coefficient depending on rope class chosen

D: nominal rope diameter

H: values are reported later in Table 7 as function of class of mechanism chosen for the rope and the object on which they are wrapped.

Rope end fittings

The end fittings elements are chosen such that they can endure a notable permanent deformation at a stress value of 2.5 times the maximum tensile stress.

The system takes into consideration the friction forces exerted on drum and on the end and must resist 2.5*S.

In the maximum unwinding position of the cable at least three completes turns must remain wrapped on drum before attaching the end of the cable. [33]

2.1.8 STANDARDS FOR ROPES

After Knowledge of Rope specifications needed, the following step will be the selection out of standards available in the CERRATO Company of the correct rope to proceed in design of other components.

Cerrato Company use traditional standards for Hoisting applications, these standards are present in Italian market and have a reputation good enough to compete in the market.

In Table 2.3 an example is given to explicit the meaning of Rope Standards and show some of its parameters.[34]

Table 2.3: Rope Standards

High resistance bright steel-Class 6*36-IWRC-ISO 2408-Wire strength 2160 N/mm2			
Construction	Ø rope mm	Weight of rope per 100 mm	Min Breaking force kgF
S9AR 6*35 WARRINGTON-IWRC Right regular lay Cod.54.150	6	15	3000
	7	21	4080
	8	27	5320
	9	34	6740
S10AR 6*36 WARRINGTON-IWRC Right regular lay Cod.54.151	10	42	8600
	11	51	10400
	12	60	12440
	13	71	14580
	14	82	16820
	15	95	19370
	16	108	22020
	17	121	24870
	18	136	27830
	19	152	31090
	20	168	34450
	21	185	37920
	22	203	41590
	23	222	45460
	24	242	49540
	25	263	53820
	26	284	58210
	28	329	67480
	30	378	77470
	32	430	88070
	34	486	97250
	36	544	109070
	38	606	121300
	40	672	134560

2.2 DRUM DESIGN

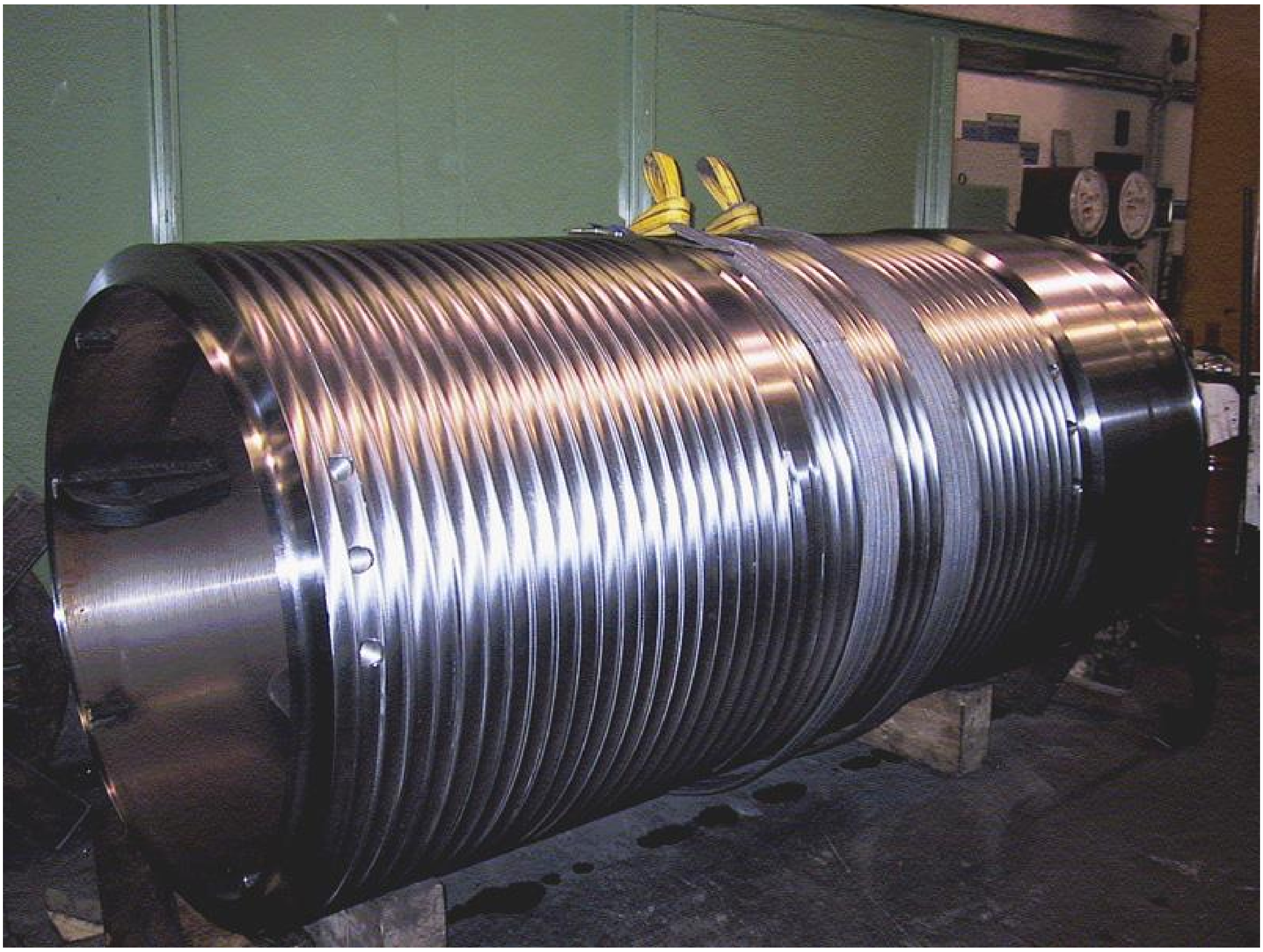

Figure 2.3: Drum shape

As shown in Figure 2.3 the inner part of the drum is empty. Therefore, the drum can be subjected to crushing and bending.[35]

Telling that the design must be performed to avoid failure by those stresses.

The crushing strength of the drum is the compressive stress that breaks the drum due to compressive forces applied on it. It is calculated from the following formula:[36]

$$\boldsymbol{Ct} = (\boldsymbol{K} \times \boldsymbol{T})/(\boldsymbol{P} \times \boldsymbol{t})$$

K constant depending on rope layers,

T tension on a single rope,

P pitch of groove,

T drum thickness.

2.2.1 DRUM PROPERTIES

The main property of the drum to design are it's diameter and it's length , it is related to the length of the ropes to be wrapped around it and to the number of ropes chosen.

Two case are present:

Case 1: Make a short drum with a big cross section (L short and D big)

This case is applied when beam width is limited. In order to compensate the short width available the drum must be made with a bigger diameter so that the sufficient length of the ropes is wrapped around the drum.

The problem in this case is that when the drum diameter increases the torque will increase:

$$(\boldsymbol{T} = \boldsymbol{D}/\mathbf{2} \times \boldsymbol{F})$$

where T is torque in N*m , F is force applied on drum in N and D is drum diameter,

Therefore limitations will also be present regarding maximum torque allowed leading also to a diameter limitation.

Case 2 : Make a long drum with a small cross section :

This case represents the complementary case for the previous one. The torque will be minimal in this case since the diameter is decreasing.

The limitations can be the beam width and can depend on size of installation zone and on beam bending capacity.[37]

Generally the drum takes a cylindrical shape and in properties determination is considered simply like a cylinder.

The drum must be grooved in a precise manner in order to be wrapped around later with the cable.

The grooving method in CERRATO Company is a traditional method used long time ago by which certain parameters are considered.

The final product will be the grooved drum with different types of diameters defined:

Minimum diameter, Drum Primitive diameter, Theoretical primitive diameter, Throat bottom Diameter.

All of these data values are shown in Chapter 3 in Program Calculations.

2.2.2 SIMULATION USING INVENTOR SOFTWARE

The company uses a version of inventor in which hoist components shapes and properties are ready to be used.

This is obvious since it saves a lot of time for engineers and it is an obligatory phase to be encountered. The CAD design has become more clear and flexible.

Figures 2.4 and 2.5 shows illustrations of the drum different sides implemented in INVENTOR software at CERRATO Company.[38]

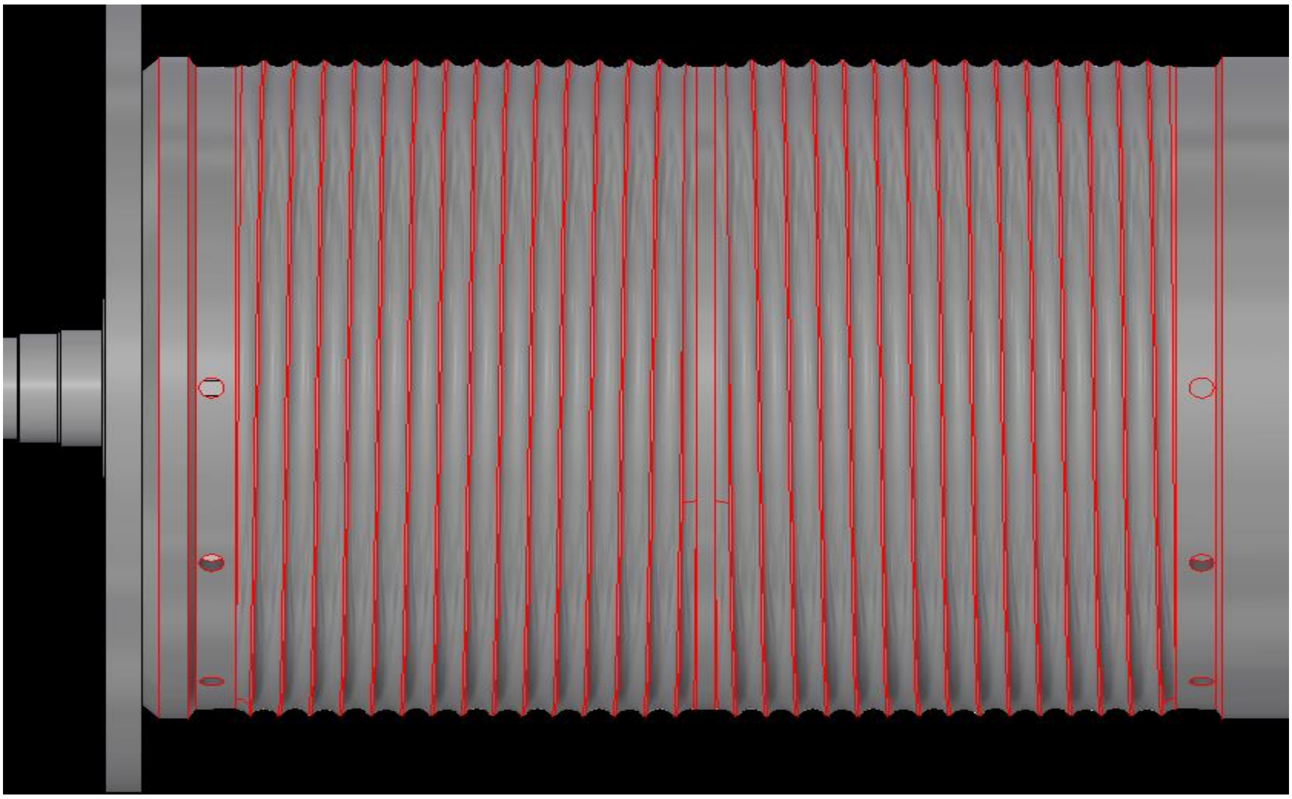

Figure 2.4: Grooved drum in inventor

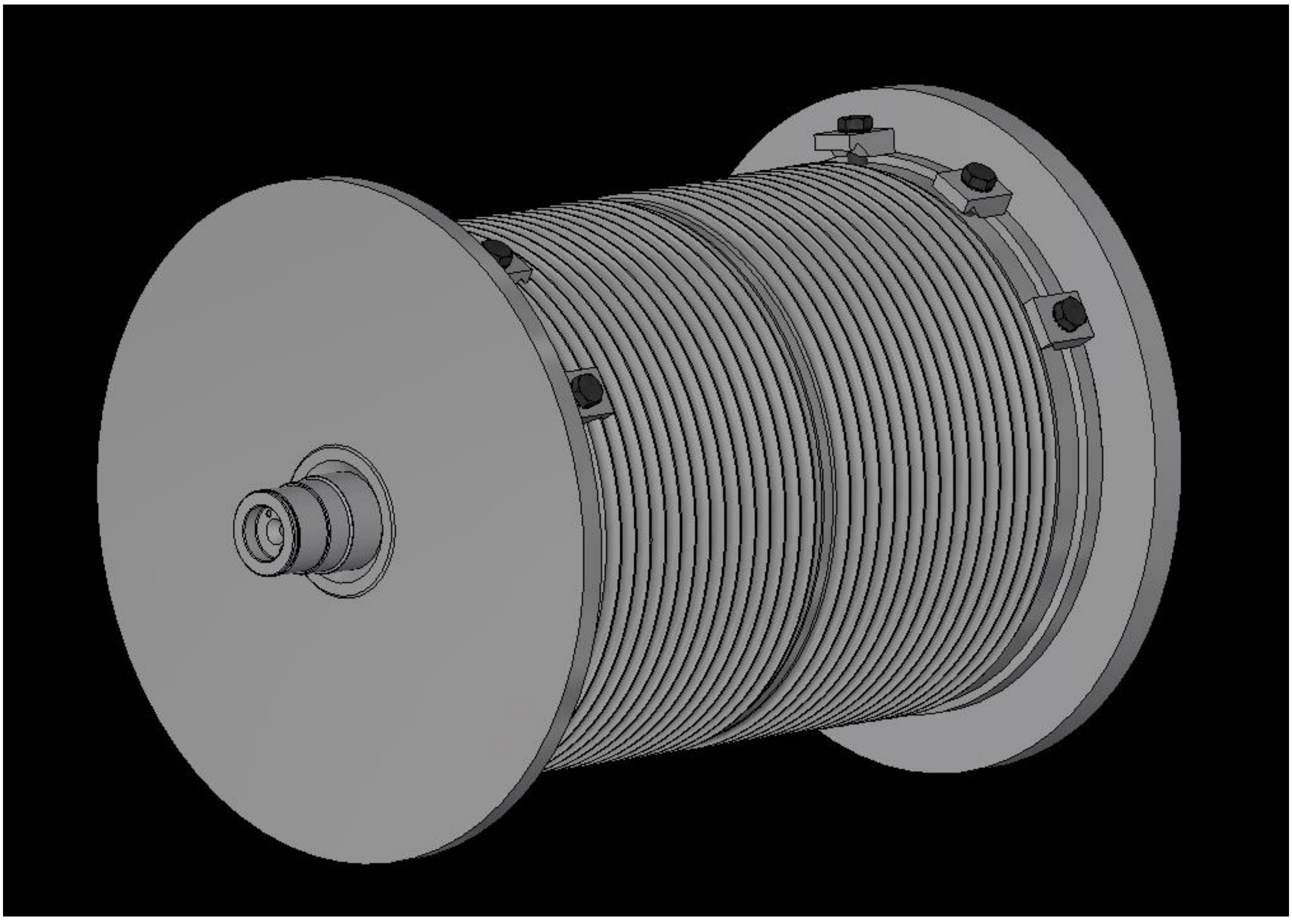

Figure 2.5: Grooved Drum side view in inventor

2.2.3 DIAMETER SELECTION

After deciding the number of pulls at the hook to be made , the total length of ropes that must be wrapped around the drum is calculated.

A safety margin is adopted to guaranty that all the rope length can be wrapped on the drum.

Note that not the whole part of the drum can be wrapped around ,the middle part of it is excluded because the ropes coming from different ends cannot meet together.

A geometrical symmetry must be maintained so the hoist system can function properly.

After selecting class of mechanism a fundamental constant H is selected to proceed for drum and pulley's diameter calculations.[39]

According to F.E.M the H values are reported in the table 2.4 .

Table 2.4: H Values as function of Classes

Class of mechanism	Drum Hd	Returning Pulley	Equilibration Pulley
M1	11.2	12.5	11.2
M2	12.5	14	12.5
M3	14	16	12.5
M4	16	18	14
M5	18	20	14
M6	20	22.4	16
M7	22.4	25	16
M8	25	28	18

DRUM

$$\boldsymbol{D = Hd \times d}$$

H values for each device is extracted from table 2.4 and the diameter is determined by simple multiplication by rope diameter.

Therefore, after determining the Rope diameter, the Drum and the returning pulleys diameters can be determined.

Figure 2.6 shows an example of 3 parallel pulleys with equal diameters already installed in Hoist system.[40]

2.2.4 PULLEY

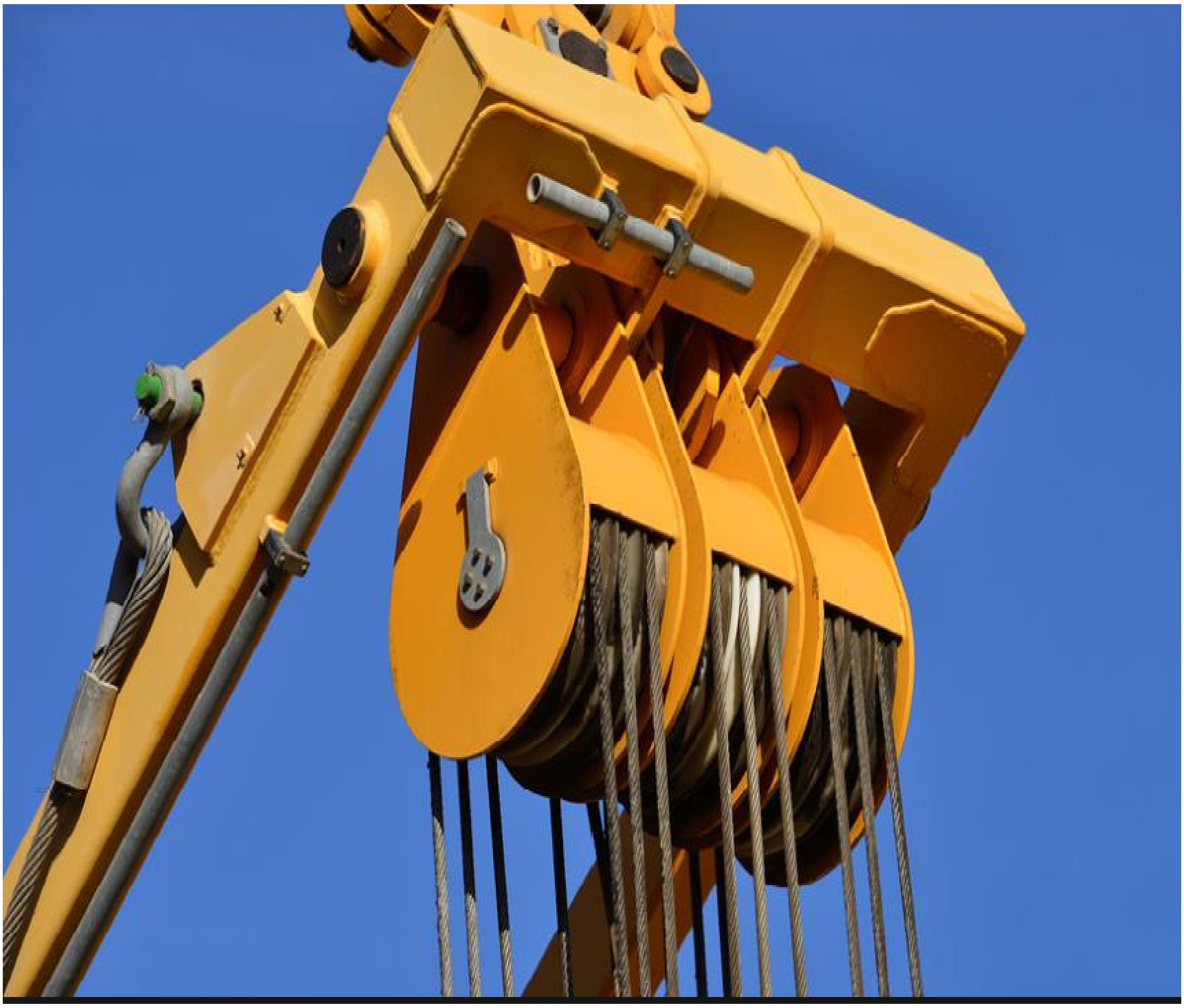

Figure 2.6: Pulleys dimensioning

Pulley selection

The pulley selection process is identical to the drum selection.

Based on case implemented, class of mechanism is selected, which will yield the constants selection Hp specific for the pulley from Table 2.4 .

Depending on pulley's type (either it is a returning pulley or an equilibration pulley).

Afterwards, The pulley diameter is determined from the following formula:

$$Dp = Hp \times d$$

2.3 SHAFT DESIGN

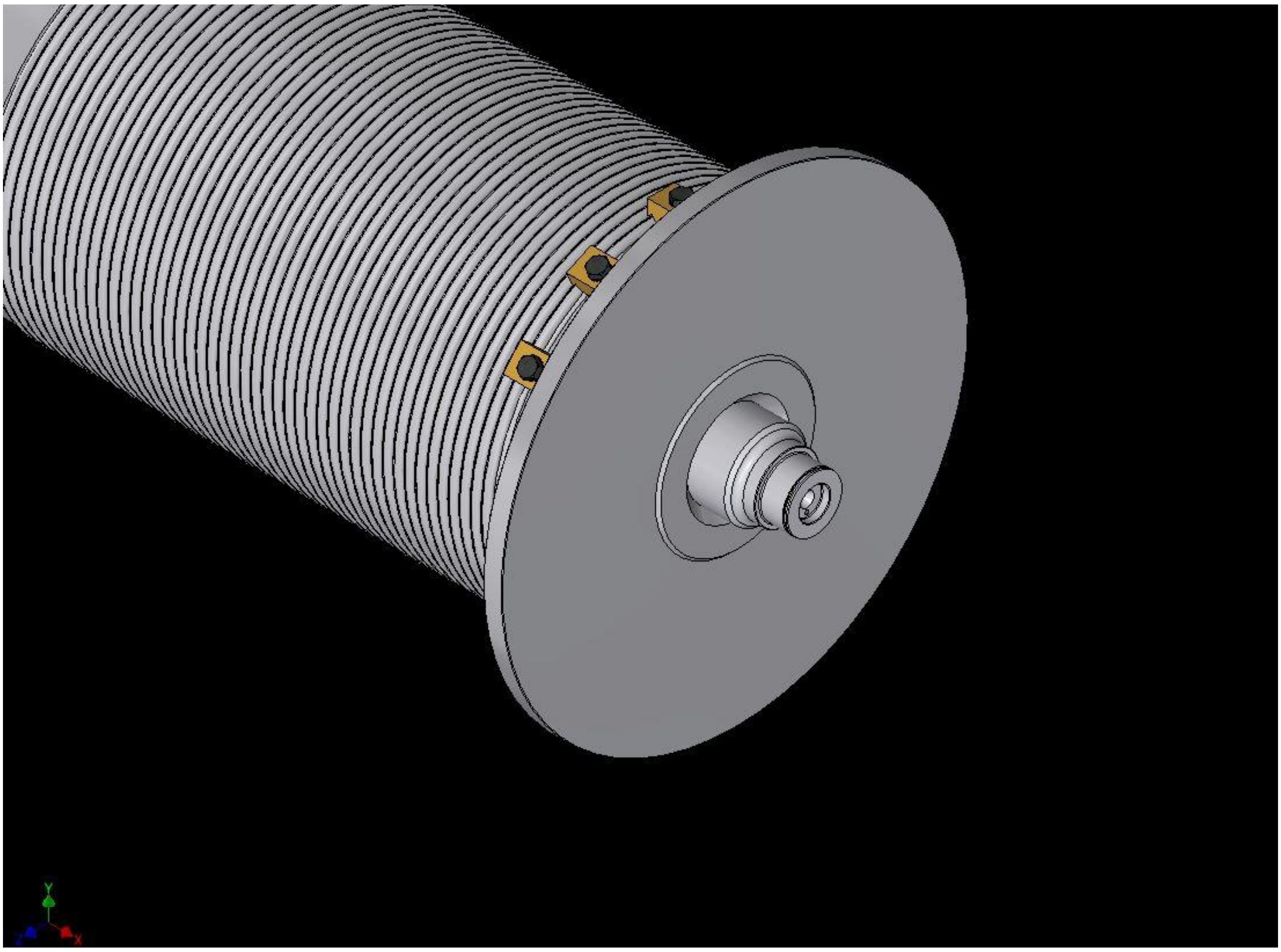

Figure 2.7: Rotating Shaft

Device used for direct transmission of motor rotation and rotating Moment.

It has the function of transmitting or receive the driving or the resistant couple.

Figure 2.7 shows an inventor simulation of a rotating shaft and drum implemented together.[41]

Axes

Device that maintain without transmission the torque .Rolling devices are free to move around it.

The main difference between the two is in load applied.

For shaft a torque is transmitted while axis is subjected to only bending loads.

Axis can be static (doesn't rotate) but can also rotate with the wheel it supports.[42]

Transmission shafts can be horizontal or vertical. A disassembled horizontal shaft is shown in Figure 2.8 .[43]

Figure 2.8: Horizontal Shaft

2.3.1 STRESSES ON SHAFT

a-Case of unique axial Flexion

Two cases are present :

1-Fixed shaft with rotating components : the dimensioning is done by assuming simple flexion with static stresses.

2-Rotating shaft and components : in this case rotational flexion is applied and dynamic stresses are considered.

b-Case of Torsion

when a shaft transmits a torsional moment along the whole rotation axis. The flexion stresses can be neglected and simple torsion case is applied.

c-Case of Flexion and Torsion

The main parameters to define are shaft material type and shaft diameter. Generally, the material type is steel able to be submitted to welding in all applications.

For the diameter determination the reference variables are the bending moment and the twisting moment.

Most of case the shaft is subjected to both torsion and flexion (not negligible).In this case the dimensioning is done using flexion-torsion criteria.[44]

The calculation for the 3 different cases are shown in the Figure 2.9 .

Formule per il dimensionamento di Alberi e Perni

Sollecitazione	*Progetto*	*Verifica*	*Significato dei simboli*
1 - Flessione	$d = \sqrt[3]{10M_f/\sigma_{am}}$ $\sigma_{am} = R/\eta$	$\sigma = M_f/W_f \leq \sigma_{am}$	d = diametro [mm] M_f = momento flettente massimo [Nmm] W_f = modulo di resistenza alla flessione [mm^3] σ_{am} = carico unitario di sicurezza normale ammissibile [N/mm^2] R = carico di rottura [N/mm^2] η = grado di sicurezza: 3 ÷ 5 per carichi statici 9 ÷ 15 per carichi dinamici
2 - Torsione	$d = \sqrt[3]{5M_t/\sigma_{am}}$ $\tau_{am} = 4/5 \cdot \sigma_{am}$	$\tau = M_t/W_t \leq \sigma_{am}$ $M_t = 9.550.000\ P/n$ $W_t \cong 0{,}2 \cdot d^3$	d = diametro teorico [mm] M_t = momento torcente [Nmm] W_t = modulo di resistenza alla torsione [Nmm] τ_{am} = carico unitario di sicurezza tangenziale ammissibile [N/mm^2] P = potenza trasmessa [kW] n = velocità di rotazione [giri/1']
3 - Flesso-Torsione	$d = \sqrt[3]{10M_{f\,id}/\sigma_i}$ $M_{f\,id} = \sqrt{M_f^2 + 0{,}75M_t^2}$	$\sigma_i = \sqrt[3]{\sigma^2 + 3\tau^2} \leq \sigma_{am}$ $\sigma = M_f/W_f$ $\tau = M_t/W_t$ $\theta = M_t \cdot l/G \cdot J_p \leq \pi/4 \cdot 180$	M_{fid} = momento flettente ideale: il momento flettente che da solo crea sull'albero lo stesso stato di pericolosità della contemporanea presenza dei momenti torcente e flettente l [mm]; G [N/mm^2]; J_p [mm^4]

Figure 2.9: Equations determining bending & twisting moments

After defining the actual situation and the proper calculation method chosen the theoretical diameter is obtained from Equations of Figure 2.9 .[45]

A safety factor must be selected for which the real diameter will surpass the theoretical one. However, it cannot be exaggerated due to fatigue effects amplification.

Afterwards, the effective diameter must be determined to take into consideration slots installation and to respect the Proportioning Normative in Transmitting shaft case.

2.3.2 SHAFT SECTION VARIATION

The shaft can be of constant section or variable.

In the drum case the shaft section is variable at extremities because of connection with bearings and gear motor. The case considered is shown in Figure 2.10 .[46]

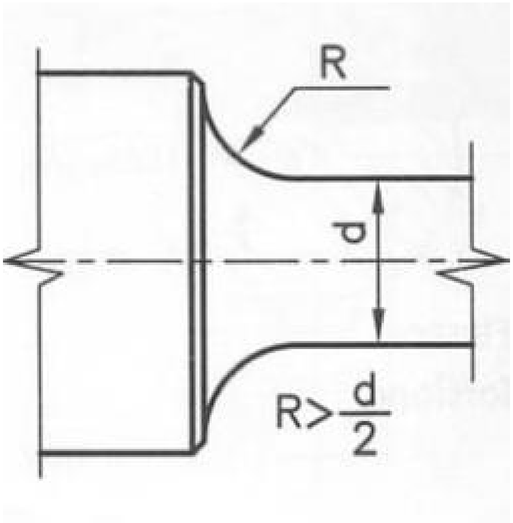

Figure 2.10: Shaft Section variation

Shaft section variation is a critical parameter for drum stability. When section variation is more relevant criticality increases due to difficulties in connecting the different parts of the system.

Many studies were made to find the relation between section variation of shaft and constraints needed to be put.

When there is a need to install bearings, bushings or supports at the extremity of the shaft,

a diameter variation less than 20 % represented in the formula below is applied :

$$\boldsymbol{(D-d)/(d \times 100) \leq 20}$$

2.3.3 SHAFT PINS

Part used to connect supports with others to allow the reciprocal rotation.

They assure system stability and prevent accidental fall of different parts like elements installed on shaft. Various types of pins exist, depending on type of load that will bear, parts to connect, and on location of installation of pins.

Pins with axial thrust:

When the pin must discharge the axial force from shaft to the supports, the pressure division must be uniform along supports.

2.3.4 PIN DIMENSIONING

A term called specific pressure "Ps" is defined representing ratio between pin supported load and it's effective section. The specific pressure must not trespass the admissible pressure. A scheme of the case is represented in Figure 2.11 .

The following formula guarantee the previous condition :

$$\boldsymbol{l > F/(Pam \times d)}$$

where F force applied , p am admissible pressure and d pin diameter.[47]

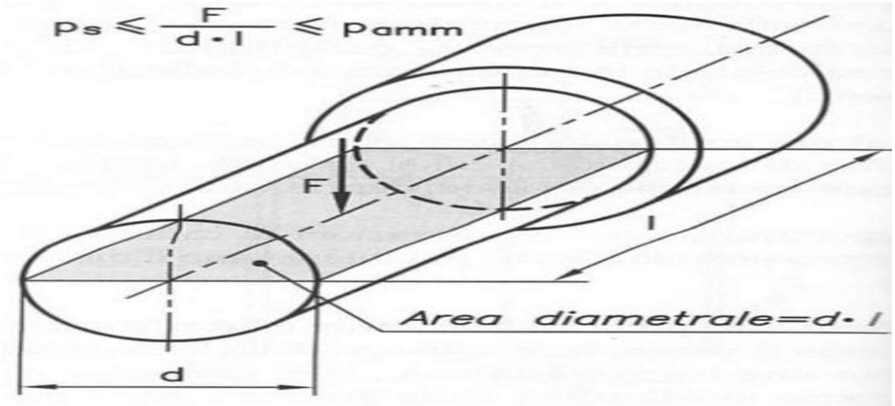

Figure 2.11: Admissible Pressure constraints

2.3.5 STEEL SHAFT WELDING OPERATIONS

Welding is a critical operation to be considered since it can create weak points. It is mainly made in variable sections zone to connect separate elements.

Steel type is an essential parameter to be known before welding starts. Working conditions and load heaviness must be also considered to define a correct safe welding operation.[48]

The following formula is used to define the weldability allowance of steel type used:

$$\boldsymbol{C\ ev = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Ni + Cu}{15}}$$

Each letter represents % of the corresponding chemical elements (as listed in periodic table).

Weldability Verification:

If $\boldsymbol{C\ ev < 0.4}$ Optimal Welding

$$\boldsymbol{If\ C\ ev\ > 0.4}$$

Precautions must be made such as heat the parts at 150 C, and use a basic type of welding arc.

2.4 SUPPORTS

Figure 2.12: Supports

Elements that sustains rotating pins in fixed position and maintaining attrition in acceptable limits. Supports have different shapes and structures as shown in Figure 2.12 .[49]

The internal part of the support can be in direct contact with the shaft or can contain an intermediate component called bearing.

2.4.1 SUPPORT TYPES

1-With frame in one piece : it has a simple construction .They have a limited use at the extremity. For shaft disassembly they must be removed.

2-With frame in two pieces : they are used when there's a need for simple disassembly of elements, over long shafts, heavy and difficult to maneuver.

The support becomes divided in two parts along the horizontal diametric section.

3-Support with oscillating bearing: used in case the shaft is subjected to deformation. In this case there's a need to follow the shaft pin oscillation so a bearing is inserted and keyed to the support to oscillate.

4-Support including lubrication devices:

They are essential in case of radial attrition. There is several types of lubricants.[50]

2.5 MOTOR

Figure 2.13: Gear Motor at CERRATO S.R.L

2.5.1 MOTOR APPLICATION

There are two types of motors installed in Crane Systems:

The first type is the unit providing the drum with rotational power leading to the trolley vertical displacement. In Figure 2.13, it is shown in black color. It is related directly to the shaft of the drum.

The second type is the one responsible for bridge and trolley translation in horizontal directions. This type is connected directly to a reducer with a self-brake. It is the one with blue color in Figure 2.13. [51]

In both cases the motor is a simple type of electric motor. It converts electrical energy into mechanical rotational energy.

However, in first case the angular speed of the gear motor is different than the angular speed of the drum.

This is due to two reasons:

1- Dimension difference of the motor shaft (inside the motor system) and the drum shaft. The diameter of the drum shaft is much larger than the motor shaft.
2- The alimentation type: the motor is directly alimented from electrical source while the shaft is only transmitting the moment the gear motor is providing.

Thus, a solution is required to resolve the following problem and it is the implementation of the gearbox independently from motor type selected.

In winch crane systems all components are installed separately for Maintenance reasons . This is why; the drum-connected motor is not related to the reducer selected.

The gear motor function is the association between the electrical motor system and the considered system. It uses the principle of converting translational velocity into rotational one.

The equilibrium of translational velocity can be converted to angular speed per diameter. This last idea presents the solution to the problem:

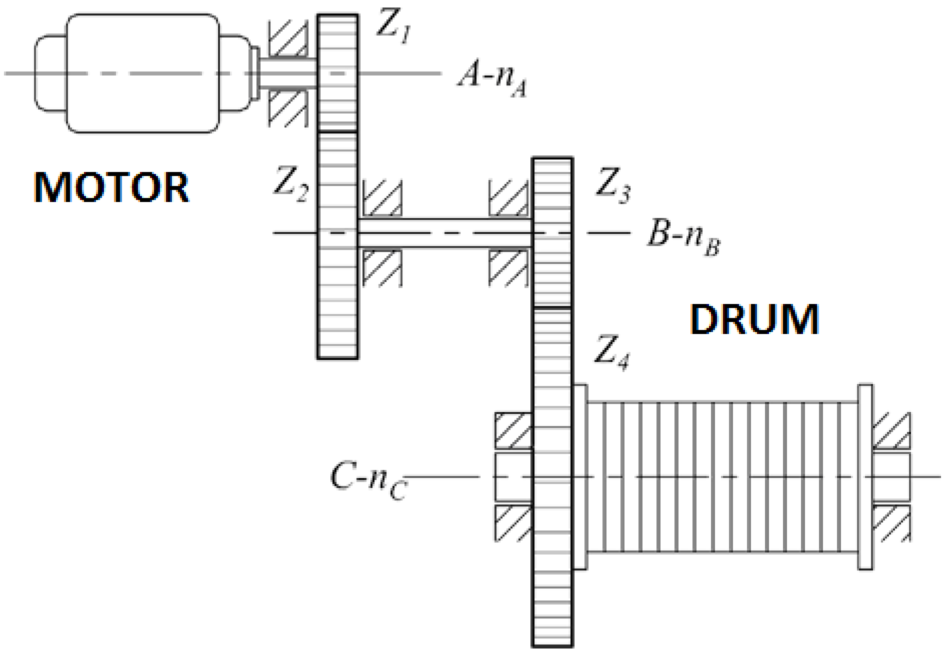

Figure 2.14: Gearbox principle

If the diameter is augmented, the angular speed of the shaft can be decreased for the same translational velocity. This is the aim because the drum shaft generally rotates slower than the electrical motor.

Thus, The electric motor rotates with a higher angular speed but with a smaller diameter while the drum shaft rotate with a lower angular speed but a bigger diameter as shown in Figure 2.14. [52]

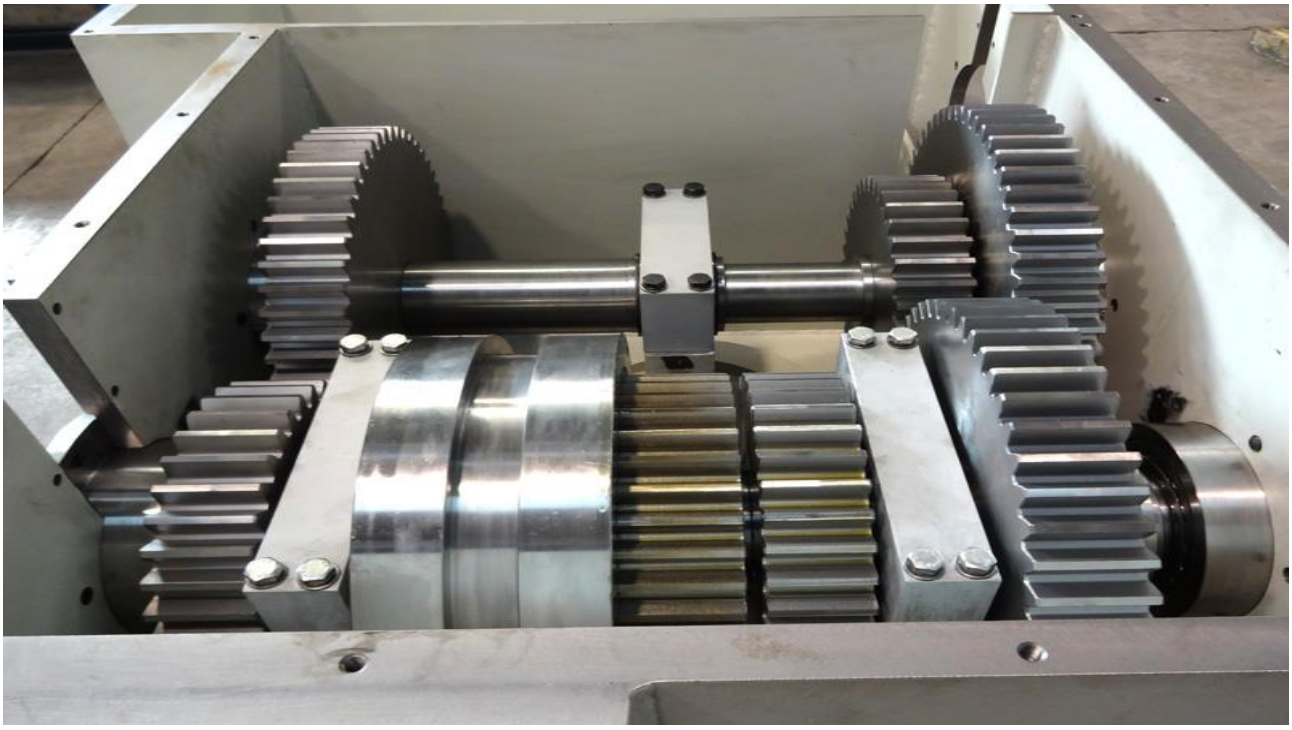

Figure 2.15: Gear pinions

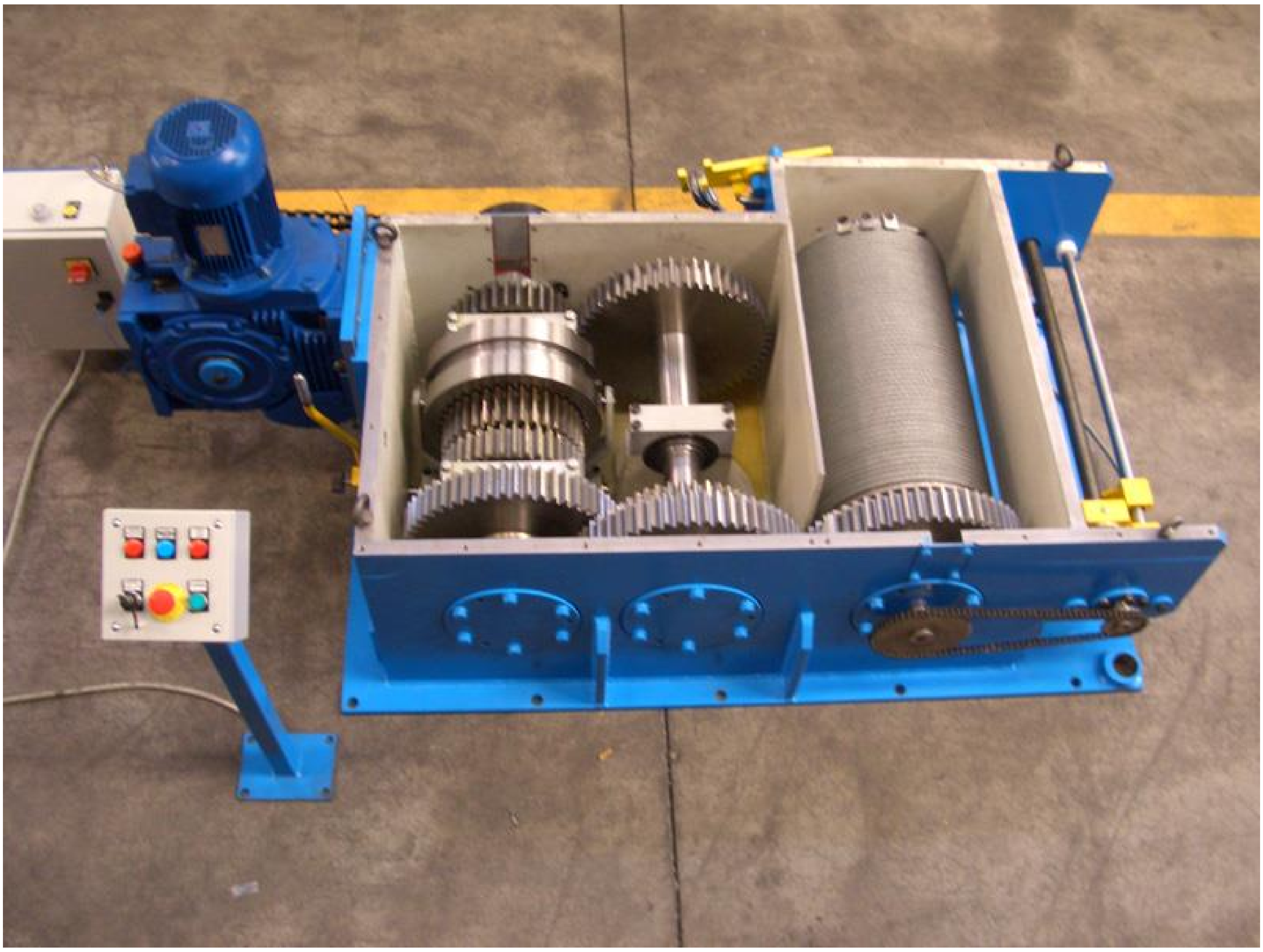

Figure 2.16: Gear motor inside crane

2.5.2 REDUCER

To Each Motor selected for Crane system a Reducer controlling the torque needed must be associated. The main properties designed of the reducer are the nominal torque and the service factor.

Reducer is mainly used to convert the torque given by motor to the one needed by crane system in order to equilibrate the system.

System continuity is assured thanks to the reducer implemented.

The reduction ratio can be variated as much as the designer want,

depending on number of pinions present. This is clearly shown in Figures 26.

In Figure 2.15 the diameter difference between rotating drum and motor shaft can be seen from the lower side of the drum casing box.[53]

2.5.3 MOTOR SELECTION METHOD

Motor selection is based on the torque needed to rotate the drum.

After the determination of the drum diameter, the torque needed for drum rotation is calculated from the following formula:

$$\boldsymbol{M = T \times D/2}$$

M torque in N*m, D Drum diameter, And T rope tension in N

Once the required torque is determined ,the required power is calculated considering drum angular speed required :

$$\boldsymbol{P = M \times \omega}$$

M torque in N*m, $\boldsymbol{\omega}$ angular speed in rad/s

It is proceeded to determine the power of motor and reducer to be chosen for the proper application from the standards available.

The Reducer selection mainly depends on Nominal torque needed and number of velocity required for the considered application.[54]

2.6 BEARINGS

Figure 2.17: Bearings Picture

Bearings are intermediately installed between support and shaft in order to :

Reduce to a minimum the attrition between rotating shaft and fixed support.

Support the transmitted force from shaft to sustaining structure.

Allow the rotation with minimal abrasion .

Helps for shaft regular rotation in place.

Depending on work application, different bearing types are defined. An illustration of bearing is shown in Figure 2.16.[55]

The most commonly used types are:

2.6.1 RADIAL BEARINGS

Figure 2.18: Radial bearing example

It is formed of a cylindrical shell that wraps the rotating shaft ,it is made of material different than of the shaft in order to reduce attrition forces. An illustration is shown in Figure 2.18.[56]

2.6.1.1 COUPLING BEARINGS-SUPPORT

The bearing can be blocked on the support by :

Screw with cup head

Screw with cylindrical head

Axial plug or screw

Bearings have different forms. In case of frame with two pieces the bearing is divided in two half's.

The right coupling between bearing and support is necessary to provide an adequate lubrication between the two.

2.6.1.2 MATERIAL FORMING BEARINGS

The material forming the bearings must have the following characteristics:

1-Low attrition coefficient

2-Good fatigue resistance

3-porous structure to hold back the lubricant

4-malleability for successful adapt with pin

5- enough hardness to support specific pressure

6-Good corrosion resistance

7-Thermal dispersion capacity

8-Silence functioning

2.6.1.3 MOST COMMON MATERIAL USED

a-Bronze

It is the most used material because it can support heavy load even in case of bad lubrication.

b-Anti-friction alloys

Alloys based on lead-antimony, tin-lead, copper, copper–lead. Used in case of heavy conditions like elevated shaft rpm and elevated unitary pressure applied.

c-Self lubricating Sintering agents

Made of microscopic structures iron based,

Their price is lower than bearing's price,

Provide a Permanent lubrication,

Encumbrances and maintenance reduction,

Have a good attrition coefficient.

d-Synthetics

Made of polyamides resins or of synthetized fibers installed with adhesive at the interior of a rigid steel support,

Can work without lubrication thus no need for maintenance,

High load resistance ,

Good dimensional stability,

High usury resistance ,

Null electrical conductivity,

High damping ability concerning vibrations.[57]

2.6.2 ROLLING BEARINGS

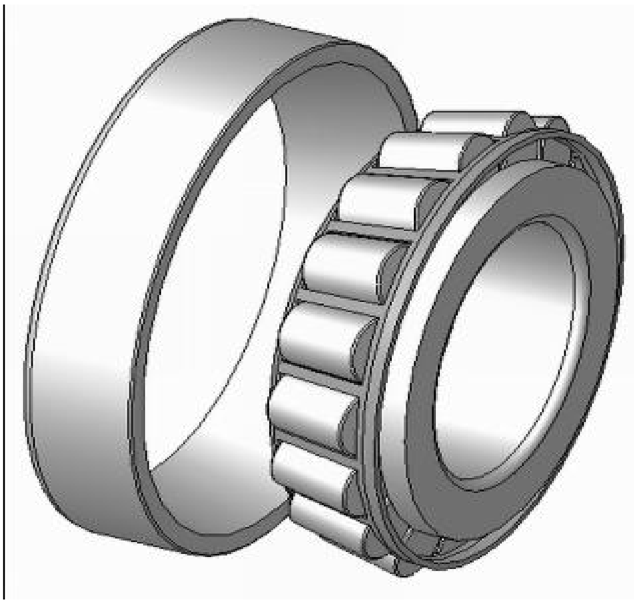

Figure 2.19: Rolling bearing example

Rolling bearing is the second common type of bearing used in mechanical industry. An example is shown in Figure 2.19.[58]

It contains rolling elements such a spheres, cylindrical rollers, conical rollers that facilitates the relative motion between rotating part and fixed part.

Compared with radial bearings rolling bearings transform radial attrition in rolling attrition with a list of advantages.

2.6.3 USING ROLLING BEARINGS INSTEAD OF RADIAL BEARINGS

Advantages

1- The attrition is 10 times lower with respect to radial bearing,
2- Lower bearing heating,
3- Abrasion reduced during functioning,
4- Lower axial encumbrance ,
5- Easy and fast changing,
6- Need of less support roughness.

Disadvantages

1- Higher cost,
2- Higher mounting problems.

Elements constituting Rolling bearings :

Inside ring, outside ring, rolling element and it's cage.

Rolling element can be a sphere, a cylinder, a cone , a barrel .

2.6.4 CLASSIFICATION OF BEARINGS

Classification is possible based on several criteria:

a-Cinematically

1- bearings for radial load (adapted to support radial loads),
2- bearings for axial load ,
3- bearing for mixed load.

b-Structurally

1- Rigid bearings ,
2- Sealed bearings,
3- Spherical bearings,
4- Bearings with groove.

c-Dimensionally

Considering Dimensional criterion three series of dimensioning are present :

1- Dimensional Series (d),
2- Diametric series (D),
3- And width series (B),

For cost considerations, calculations, quality required, and flexibility the number of commercial bearings is limited to the ISO's choice :

1- For radial bearing the ISO 15,
2- For conical roller the ISO 355,
3- For axial bearing the ISO 104.

2.6.5 CHOICE OF BEARINGS

The choice depends upon a lot of criteria and requires experience to choose the right type.[59]

Below is listed some of them :

1- Space available,
2- Load direction,
3- Specification of rotating system,
4- Work velocity,
5- Silence level required,
6- Axial movement,
7- Assembly and disassembly,
8- Protection and lubrication of rolling element.

2.6.6.1 LUBRICATION OF ROLLING BEARINGS

1- Reduce the attrition of rotating elements,
2- Protect from oxidation,
3- Remove heat developed,
4- Silence functioning,
5- Minor vibration ,
6- Hinder dust entrance a right lubrication allows a good functioning and extends it’s service life reducing it’s abrasion.

The lubrication choice depends on work conditions such as :

1- Temperature
2- Velocity,
3- Maintenance frequency,

Effect of surrounding ambient and circumstances.

It can be :

Grease lubrication : used when oil use is inconvenient or difficult,

Oil lubrication : better than grease lubrication , preferred in case of central or forced lubrication , in this case the control of lubricant level is easy.[60]

A schematic of Bearing Lubrication is shown in Figure 2.20.[61]

2.6.6.2 LUBRICATION SYSTEM

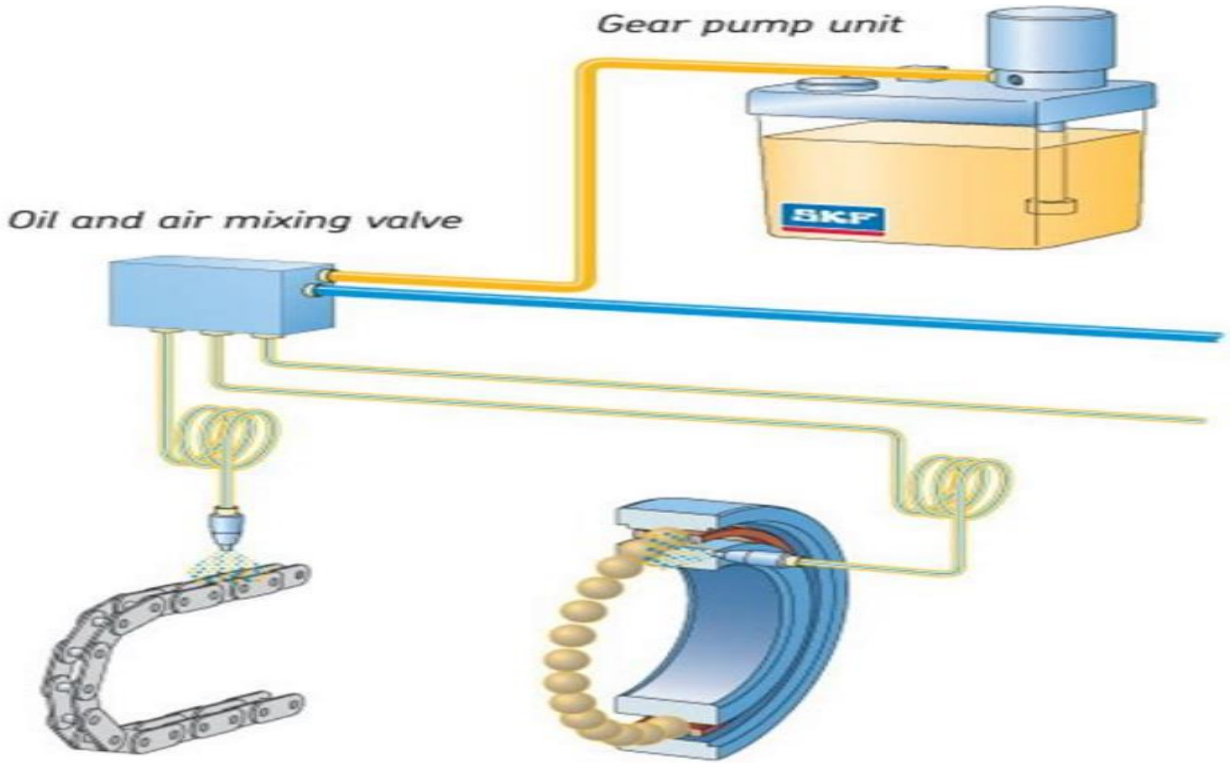

Figure 2.20: Schematic of Lubrication system

2.6.6.3 LUBRICATION TYPES

a-Manual

The lubricant is added manually. This type is the Lubrication employed in CERRATO company.

Some other types are defined as “long life bearings” which means the Bearing is not lubricated during working because it does not need lubrication in its whole lifetime.

b-Immersion

Used only in case of oil being the lubricant , the bearing is partially covered with oil .

c-Forced circulation

Used in cases where heat evacuation is necessary to avoid pump overheating and oil radiator.

d-Oil jet

Used for bearing with high velocity .

2.7 GEAR COUPLINGS

Figure 2.21: Gear Couplings

In the Crane system case, the gear couplings function is to assure connection between gearbox (motor reducer) and drum shaft during rotation.

The Gear Couplings must be flexible to compensate misalignments of shafts to be connected.

The main technical data for selection of Gear Coupling is the nominal torque and the carrying load.

Since its function is to assure synchronous rotation between reducer and shaft it must meet the system shaft/reducer requirements to afford the best working condition.

The Joint constraint is to be able to support a torque (nominal torque) higher than the one for shaft/reducer.

A verification related to carried load is also made to assure the couplings can bear the mounted load.

In case the Couplings diameter is not enough for the application larger diameters are examined until the couplings meets the required conditions.[62]

A Gear coupling example is shown in Figure 2.21.[63]

CHAPTER 3: DESIGN PROGRAM

3.1 CALCULATION RESULTS

The Design Program is split and described in Two parts.

In section one, all the parameters are defined and assigned real values in order to determine the main Design data. Parameters are imported from Tables, Special Curves & Catalogs or calculated in Section2.[64]

For all parameters is assigned a name, symbol, value, unit of measurement and equation if present. The data is divided in different parts depending on Component studied and type of data:

1- Commercial Data Input:
   This type of data is an input given by Client upon his needs and applications. Client can assign values, or can precise objective of Crane System installation and permitting CERRATO consultancy team to suggest empirical values of data based on Company design history.
   Commercial Data Results of the case are shown in Table 8.
2- Technical Input Data:
   This data is mainly an input of engineers/technicians in CERRATO Company. Values are assigned based on eligible design and experience of Technical Staff in Technical office. Catalogs and normative are main source of information from which this data type can be extracted.
   Technical Data Results of the case are shown in Table 9.
3- Rope Data Assignment:
   This part mainly defines fundamental ropes parameters to be determined. The essential thing to determine for the selected rope is its diameter and its material type.
   If breakage resistance is okay, it is proceeded to choose rope diameter to assure required load tolerance is possible with a safety margin specified in safety factor.
   A series of verification will be made to ensure the validation of diameter and resistance to breakage.
   Rope Data Results of the case are shown in Table 10.
4- Drum Data Assignment:
   This part defines fundamental drum and drum/system parameters.
   The main criteria to check for drum is its class of mechanism and its lifting speed so that it meets the requirements needed. The diameter must meet the rope-drum specifications needed.
   Drum Data Results of the case are shown in Table 11.
   Verifications: A list of verifications in all parts is made to ensure safe working conditions .

Calculation procedure

In section two, everything related to calculations made in program are shown. All formulas used to determine parameters and make verifications are shown clearly. The values of the different parameters are shown in section one. This section shows in detail the conducted procedure to the determination of all useful parameters calculated in program. There are some used in the formulas previously defined in section 1. For Clearer Understanding, Refer to Section 1 Results.[65]

Calculation Results

All Results of the Design are shown in Tables below:

Table 3.1: COMMERCIAL DATA INPUT

Input Data				
Characteristics	**Symbol**	**Value**	**Unit**	**Equation**
Load	P	33000	kg	
width	S	6	m	
Lift velocity	V_{soll}	3,5	m/min	
Trolley translational velocity	$V_{t\text{-}carrello}$	20	m/min	
bridge translational velocity	$V_{t\text{-}ponte}$	40	m/min	
Hook range	c	6	m	
Voltage of alimentation		400	V	
Frequency of alimentation		50	Hz	
Structural Class	A	7		
Class of Mechanism	M	7		

Table 3.2: TECHNICAL DATA INPUT

INPUT PARAMETERS ROPE-DRUM				
CARACTERISTICS	**SYMBOL**	**VALUE**	**Unit**	**MINIMAL VALUE**
Rope diameter	d	19	mm	14,6
Minimal breakage resistence	R_0	2150	MPa	
Filling factor	f	0,59		
Minimum calculated break stress	Y	0,8		
N° drums/N° pull per each drum/N° pull at hook	combo	1x2x8		
N° principal drums	$N_{princ\text{-}tamb}$	1		
External diameter for raw tube	D_e	457,2	mm	216
Raw tube thickness	S_{tubo}	25	mm	
Distance between central ropes (0 per 1 tiro dal tamburo)	x	200	mm	168
Distance between real supports	i	2200	mm	659
Shaft length till bearing center	a	500	mm	259
Distance drum axis/block pulley	b	1500	mm	
Distance drum axis/fixed pulley	c	5000	mm	

Table 3.3: ROPE DATA

VERIFICATIONS ROPE				
CARACTERISTICS	SYMBOL	VALUE	Unit	MINIMAL VALUE
Class of mechanisms	M	7		
Load to hook (add suspended masses)	P	33000	kg	
No. of drums	T	1		
N ° shots from each drum	$N_{tiri-tamb}$	2		
No. of hooks	$N_{tiri-gancio}$	8		
Coefficient ZP prospect 1 ISO 4308	zp	7,1		
Pulling force on the rope	F_0	40466	N	
Real tensile load on the rope to be chosen in the catalog	F_{reale}	287310	N	
Minimum rope diameter	d_0	19	mm	
Cable diameter check	$d>d_0$	VERIFICATO	107%	

VERIFICATION DRUM

CARACTERISTICS	SYMBOL	VALUE	Unit	MINIMAL VALUE
Class of mechanisms	M	7		
Hook travel	c	6000	mm	
Drum material	mat	S355JR (Fe510B)		
Lifting speed	V_{soll}	3,5	m/min	
mm processing raw tube on the diameter	lav	6	mm	
Tool diameter for machining	$2xr_g$	20,4	mm	
Coefficient h1 table 2 ISO 4308	h_1	22		
Coefficient h2 prospect 2 ISO 4308	h_2	25		
Coefficient h3 table 2 ISO 4308	h_3	16		
Rope pitch	p	21,5	mm	
Minimum drum diameter	D_1	426	mm	
Minimum diameter for transmission pulleys	D_2	475	mm	
Minimum diameter for balancing pulleys	D_3	304	mm	
Diameter ridges	D_c	451	mm	
Primitive theoretical diameter	$D_{pteorico}$	453,6	mm	
Primitive diameter rounded down	D_p	453	mm	
Inner tube diameter	D_i	407,2	mm	
Throat bottom diameter	D_g	434	mm	
Thickness of tube under the ring	s	13	mm	
Single drum section resistance module	W_g	1806148	mm^3	
Number of rope sections for each single rope pull from the drum	X	4		
Indicative media distance single drum	L	1425	mm	
Horizontal excursion of a rope on the single drum	L_1	363	mm	
N ° of drum turns needed a rope on the single drum	N_s	16,86		
Indicative drum length without shaft	L_{tamb}	1700	mm	
Deflection angle movable pulley (block)	α_b	2,77	gradi	
Fixed pulley deviation angle	α_f	2,08	gradi	
Maximum deviation angle	α_{amm}	3,00	gradi	
Check angle deflection mobile pulley (block)	$\alpha_b<\alpha_{amm}$	VERIFICATO	62%	
Check angle of fixed pulley deviation	$\alpha_f<\alpha_{amm}$	VERIFICATO	46%	
Indicative drum weight	m	586	kg	
Number of drum revolutions	n	9,84	giri/min	
Operating hours according to class M	t	6300	h	
Total number of cycles	N	3718534	cicli	
Static admissible tension	σ_{amm-st}	240	MPa	
Permissible tension fatigue 2x106	$\sigma_d 2x10^6$	160	MPa	
Fatigue allowable stress 5x106	$\sigma_d 5x10^6$	118	MPa	

Table 3.4: DRUM DATA

Admissible tension fatigue 1x108	$\sigma_d 1x10^8$	65	MPa	
Admissible stress fatigue	σ_d	130	MPa	
Minimum allowable tension	σ_{amm}	130	MPa	
Local compression tension A	σ_{cA}	70,2	MPa	
Local compression tension B	σ_{cB}	119,4	MPa	
Local bending tension A-A	σ_{f1}	38	MPa	
Global bending tension A-A	σ_{fg}	22,4	MPa	
Global ideal tension A	σ_{idA}	95,7	MPa	
Local ideal tension B	σ_{idB}	113,3	MPa	
Verification of ideal global static tension A	$\sigma_{idA}<\sigma_{amm}$	VERIFICATO	49%	
Ideal static local tension test B	$\sigma_{idB}<\sigma_{amm}$	VERIFICATO	64%	
Verification of the ideal ideal global tension A	A	VERIFICATO	17%	
Check the ideal local fatigue tension B	B	VERIFICATO	41%	
Safety factor of the drum joint	SF	1,8		
Nominal torque to the drum joint	C_{nom}	3364	kgm	
Radial load to the drum joint	$F_{radiale}$	4418	kg	
Joint Mania size	Mania	260		

3.2 CALCULATION PROCEDURE

3.2.1 ROPE DATA CALCULATIONS

1- Tension force on one rope is determined:

$$F0 = \frac{(P \times g)}{Ng}$$

2- Real force acting on rope Calculation :

$$Fr = F0 \times Zp$$

3- Minimum Rope diameter :

$$dmin = \sqrt{(4 \times Zp \times F0)/\sqrt{}(\pi \times f \times \gamma)}$$

4- Verification of Rope diameter :

$$d > d0$$

3.2.2 DRUM DATA CALCULATIONS

1- Diameter for machining :

$$2 \times rg = 2 \times 0.5375 \times d$$

2- Minimum Drum Diameter:

$$D1 = h1 \times d$$

3- Minimum transmission pulley diameter:

$$D2 = h2 \times d$$

4- Minimum balancing pulley diameter :

$$D3 = h3 \times d$$

5- Diameter ridges :

$$Dc = De - lav$$

6- Theoretical primitive diameter:

$$Dp\ theo = Dc + 0.125 \times d$$

7- Inner tube diameter:

$$Di = De - 2 \times s\ tubo$$

8- Throat bottom diameter:

$$Dg = Dp - d$$

9- Tube thickness under ring:

$$s = (Dg - Di)/2$$

10- Resistance module of drum section :

$$Wg = \pi \times \left(Dg^4 - Di^4\right)/(32 \times Dg)$$

11- Indicative distance for single drum:

$$L = Nt \times \left(c \times X/(Dp \times \pi)\right) \times p + x + a$$

12- Horizontal excursion of rope :

$$L1 = \left(c \times X/(Dp \times \pi)\right) \times p$$

13- Number of drum turns needed for rope:

$$Ns = \frac{L1}{p}$$

14- Drum length without shaft:

$$L\ tamb = i - a$$

15- Deflection angle for moveable pulley:

$$\alpha b = \tan^{-1}\left(L1/(c + b)\right)$$

16- Fixed Pulley deviation angle:

$$\boldsymbol{\alpha f} = \tan^{-1}(\boldsymbol{L1}/(2\boldsymbol{c}))$$

17- Mobile Pulley angle Verification:

$$\boldsymbol{\alpha b} < \boldsymbol{\alpha\ adm}$$

18- Fixed Pulley angle Verification :

$$\boldsymbol{\alpha f} < \boldsymbol{\alpha\ adm}$$

19- Drum weight:

$$\boldsymbol{m} = \left(\left(\frac{\boldsymbol{De}}{2}\right)^2 - \left(\frac{\boldsymbol{Di}}{2}\right)^2\right) \times \left(\boldsymbol{\pi} \times \frac{\frac{\boldsymbol{i}}{7.85}}{10^6}\right)$$

20- Number of drum revolutions:

$$\boldsymbol{n} = \boldsymbol{Vsol} \times 10^3 \times \boldsymbol{Ng}/(\boldsymbol{T} \times \boldsymbol{Nt} \times \boldsymbol{Dp} \times \boldsymbol{\pi})$$

21- Local compression tension at point A:

$$\boldsymbol{\sigma cA} = 1/2 \times \boldsymbol{Fo}/(\boldsymbol{P} \times \boldsymbol{S})$$

22- Local compression tension at point B:

$$\boldsymbol{\sigma cB} = 0.85 \times \boldsymbol{Fo}/(\boldsymbol{P} \times \boldsymbol{S})$$

23- Local bending tension at section A-A:

$$\boldsymbol{\sigma f1} = 0.96 \times \boldsymbol{Fo}/\sqrt{(\boldsymbol{s}^3 \times \boldsymbol{Dg})}$$

24- Global bending tension at section A-A:

$$\boldsymbol{\sigma fg} = \boldsymbol{F0} \times (\boldsymbol{i} - \boldsymbol{x})/(2 \times \boldsymbol{Wg})$$

25- Global ideal tension at point A:

$$\boldsymbol{\sigma idA} = \sqrt{\left(\boldsymbol{\sigma cA}^2 + (\boldsymbol{\sigma f1} + \boldsymbol{\sigma fg})^2 + \boldsymbol{\sigma cA} \times (\boldsymbol{\sigma f1} + \boldsymbol{\sigma fg})\right)}$$

26- Local ideal tension at point B:

$$\boldsymbol{\sigma idB} = \boldsymbol{\sigma cB}$$

27- Verification of ideal global static tension at point A:

$$\boldsymbol{\sigma idA} < \boldsymbol{\sigma adm}$$

28- Verification of ideal local static tension at point B:

$$\boldsymbol{\sigma idB} < \boldsymbol{\sigma adm}$$

29- Nominal torque exerted on joint:

$$\boldsymbol{Cnom} = (\boldsymbol{SF} \times \boldsymbol{Dp} \times \boldsymbol{P} \times \boldsymbol{Nt})/(2000 \times \boldsymbol{Ng} \times \boldsymbol{T})$$

30- Radial force exerted on joint:

$$\boldsymbol{F\ radiale} = 1/2 \times \left(\boldsymbol{F0} \times \frac{\boldsymbol{Nt}}{\boldsymbol{g}} + \boldsymbol{m}\right)$$

3.2.3 SHAFT DATA CALCULATIONS

1- Total number of shaft rotations:

$$\boldsymbol{nt = n \times h \times 60}$$

2- Diameter variation along sections:

$$\frac{\boldsymbol{D}}{\boldsymbol{d}} = \frac{\boldsymbol{Dm}}{\boldsymbol{D}}$$

3- r/d equivalente:

$$\frac{\boldsymbol{r}}{\boldsymbol{d}} = \left(\frac{\boldsymbol{Dm}}{\boldsymbol{D}}\right) + \boldsymbol{q}$$

4- area:

$$\boldsymbol{A = \pi \times ((D/2)^2 - (Di/2)^2))}$$

5- Flexional Resistance Modulus:

$$\boldsymbol{W = \pi \times 32 \times (D^4 - Di^4)/(D \times 1000)}$$

6- Torsional Resistance Modulus:

$$\boldsymbol{Wt = 2 \times W}$$

3.2.4 BENDING MOMENT & SHEAR FORCE CALCULATIONS

1- Vertical flexion:

$$\boldsymbol{Mv = a \times \frac{V}{1000} \times 1.15 \times \gamma}$$

2- Vertical shear:

$$\boldsymbol{Tv = V \times 1.15 \times \gamma}$$

3.2.5 ADMISSIBLE STRESSES CALCULATIONS

1- First admissible tensional stress :

$$\boldsymbol{\sigma A1 = \frac{\sigma R}{2.2}}$$

2- Third admissible tensional stress:

$$\boldsymbol{\sigma A3 = \frac{\sigma R}{1.8}}$$

3- First admissible shear stress:

$$\boldsymbol{\tau A1 = \sigma A1/\sqrt{3}}$$

4- Third admissible shear stress:

$$: \boldsymbol{\tau A3 = \sigma A3/\sqrt{3}}$$

3.2.6 BREAKAGE RESISTANCE CALCULATIONS

1- Tensional stress:

$$\boldsymbol{\sigma A1 = \frac{Mv}{W}}$$

2- Shear stress:

$$\boldsymbol{\tau v = \frac{Tv}{A} \times \frac{4}{3}}$$

3- Ideal stress:

$$\boldsymbol{\sigma idl = \sigma v}$$

3.2.7 ROPE SUSPENSION ANGLE CALCULATIONS

On the right side of the figure the rope is represented from different sides.

The aim of these different perspective illustrations is to clarify the presence of different types of solicitations on the rope section.

The first picture represents the rope wrapped around the drum. It can be subjected to sliding of the groove.

The second one shows how the rope will be wrapped around grooved drum. It can be subjected to erosion due to high friction effects leading to rope resistance section weakening and maybe rope cut.

The last one is showing the inclined suspension of the rope from upside of drum to the bottom side of hook.

If inappropriate angles are reached the rope will be cut. For this reason, a threshold for deviation angle is put to assure Rope safety. In the thesis case the threshold put is 2 degrees.

The variation of this angle as function of hook travel is plotted in Figure 3.1. [66]

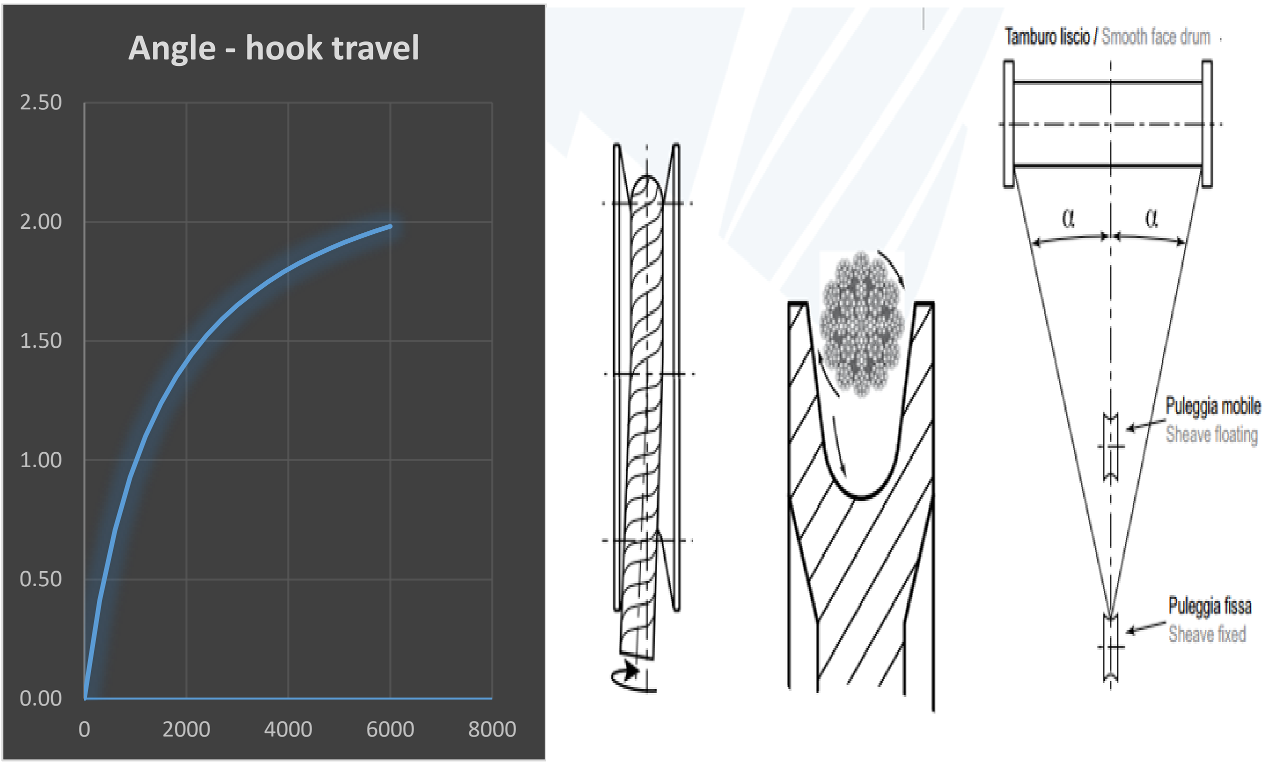

Figure 3.1: Schematic showing hook travel in real case

CHAPTER 4: FATIGUE ANALYSIS

4.1 GENERAL DEFINITION

Fatigue of a mechanical element is the decrease of performance of the element as the absolute functioning time of the element is increasing.

This decline in performance can lead to the failure of the element. Therefore, for system safety a fatigue analysis is made from which the lifetime of the element is determined.

After surpassing intended lifetime, the element must be replaced with a new one.

The fatigue resistance of an element is determined by taking into consideration:

-The material from which it is constructed.

-form, surface shape, corrosion state, dimensions and other factor that creates concentrated stresses.

-the stress ratio R representing the ratio between minimal and maximal stress for which the element will be subjected to during various load cycles.

-Number of stress cycles.

The departure point is the endurance limit against traction and alternated stresses (R=-1).

The decrease in fatigue resistance because of, the geometrical form of the piece, it's surface shape, it's corrosive state and dimensions is considered by the introduction of the adequate factors.

The endurance limit determined, for a defined stress ratio is the base for the construction of Wohler curve.

After determination of Wohler curve (fatigue resistance under the action of stress cycles having same R ratio between extreme conditions),

the fatigue resistance can be determined as function of the class of the element.

The method that will be described to find the fatigue resistance is only valid for the case of homogeneous elements.

All the considered section must be made of same material. It is not valid for treated elements.

The section studied is of cylindrical shape since the shaft is cylindrical. The forces present are traction and compression forces.

All new useful parameters $Fmax, Fmin, \sigma max, \sigma \min, R, A$ are introduced in the Figure 4.1.[67]

4.2 GENERAL CASE

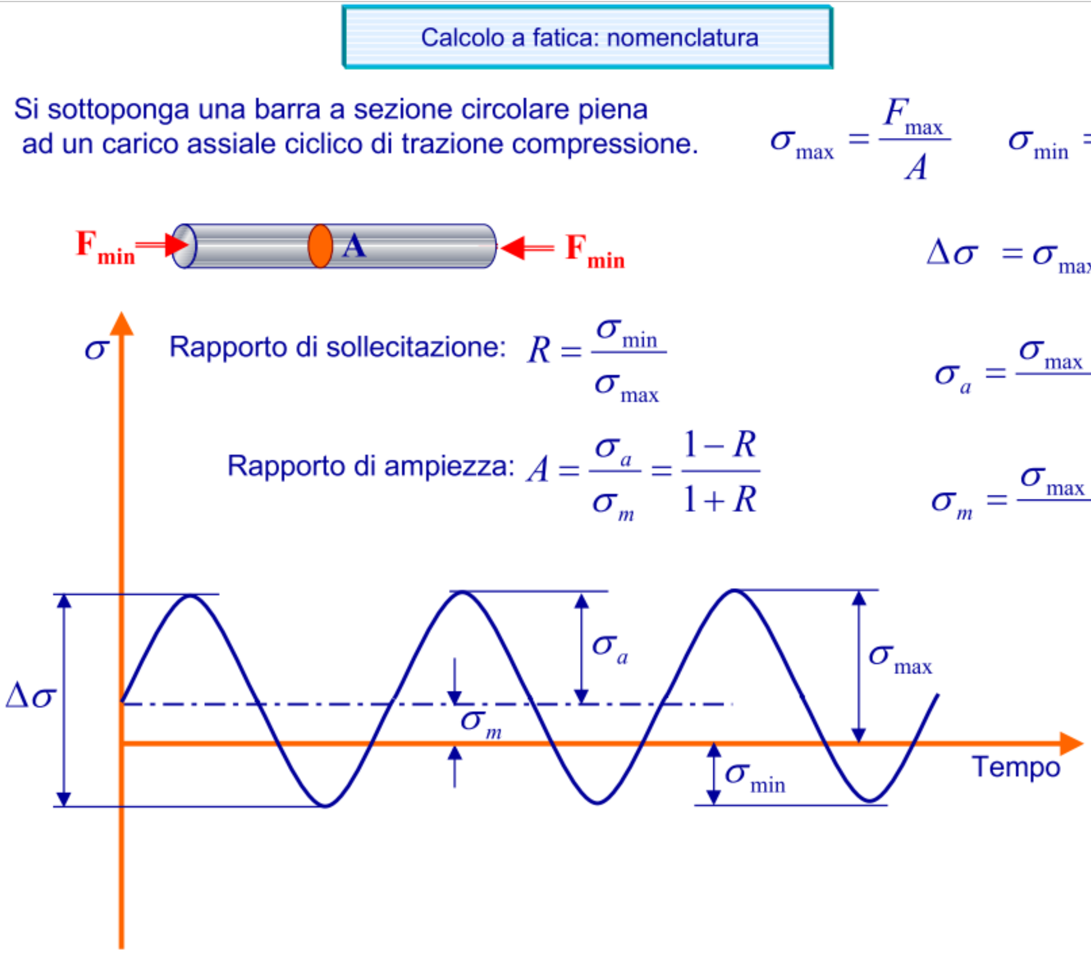

Figure 4.1: General curve

In Figure 4.1 new parameters are introduced to start the fatigue analysis.

Stress ratio R: it is the ratio between the minimal stress to the maximal stress, both of them are measured during whole lifetime of the element.

F min is the minimal force on element while F max is the maximal force,

F min gives σmin and F max gives σmax

$\boldsymbol{\sigma a}$ is called alternated stress, $\boldsymbol{\sigma m}$ is the median stress,

Both of them are defined in Figure 4.1 and they are to be used later in fatigue analysis.

$\Delta\boldsymbol{\sigma}$ is the stress difference between maximum stress and minimum stress as shown in Figure 4.1.

Two situations are implemented in Industry regarding fatigue analysis:

$$R = 0 \;\; Unidirectional\ Stress$$

$$R = -1\ Alternated\ stress$$

4.3 CASE STUDY

A-Unidirectional stress R=0

The shaft remains static and attached elements rotates. Every loading phase produces a fatigue cycle; this case presents less fatigue than the second case. In this case, the different parameters for this particular case are seen in Figure 4.2.

This case is not prevailing because of difficulties encountered mainly automatization of rotating components.

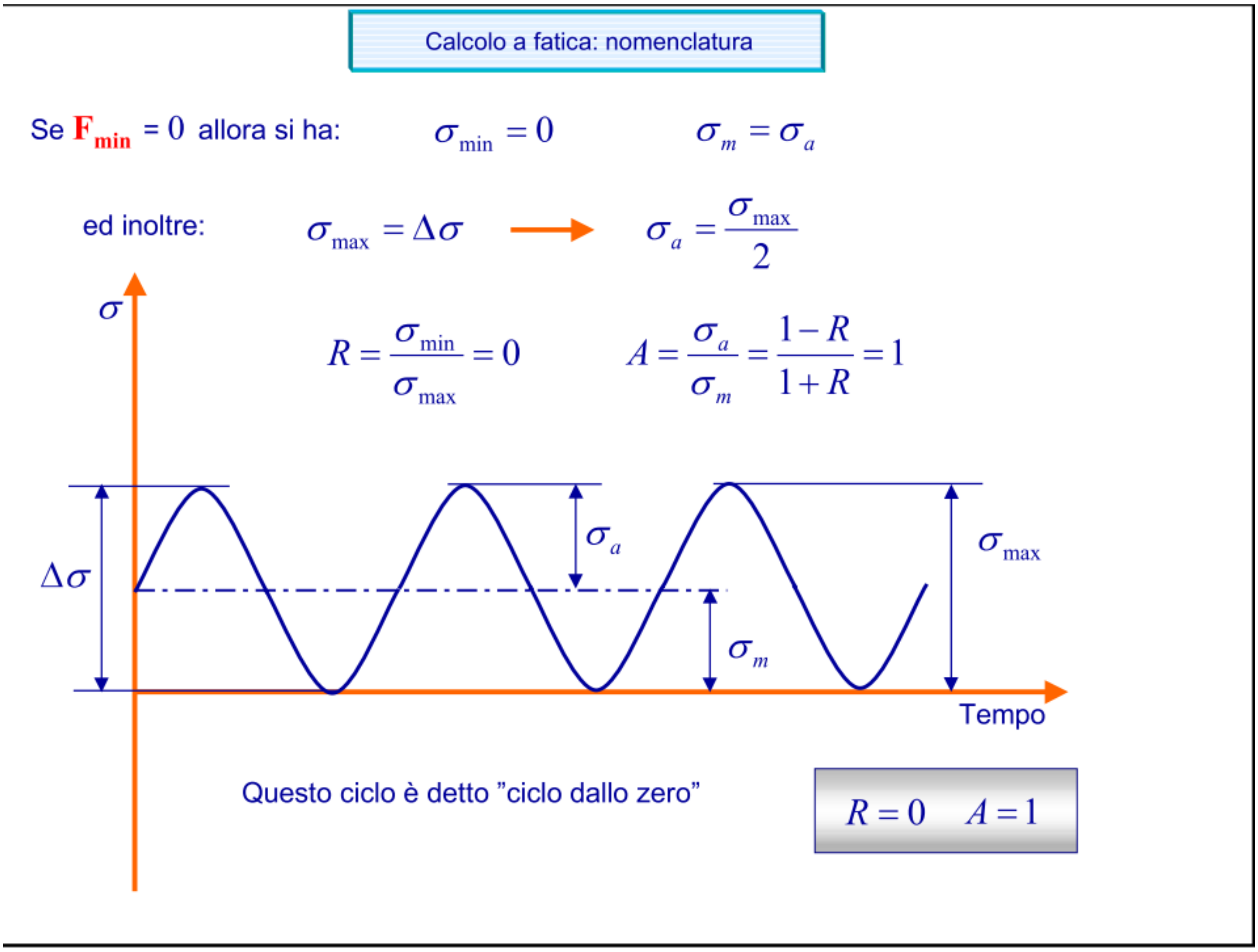

Figure 4.2: Unidirectional stress fatigue

B-Fully Reversed stress R=-1

This case is the most common case for CERRATO s.r.l Company. It is the extreme case regarding fatigue. Every point of the shaft is alternatively subjected to tension and compression each half turn.

Nevertheless, it is more common because it does not present the problem of automatization like case one.

Every half turn for each fixed point on the rotating shaft the solicitations is inverted from tension to compression and vice versa.

This case is more common because the shaft is rotating while the components are fixed on it. At this point it can be said that stress ratio R is -1 (Fmin=-Fmax).It is called fully reversed case.

For every turn of shaft, a fatigue cycle is made. This case parameters are represented in Figure 4.3.

4.4 WÖHLER CURVE METHOD

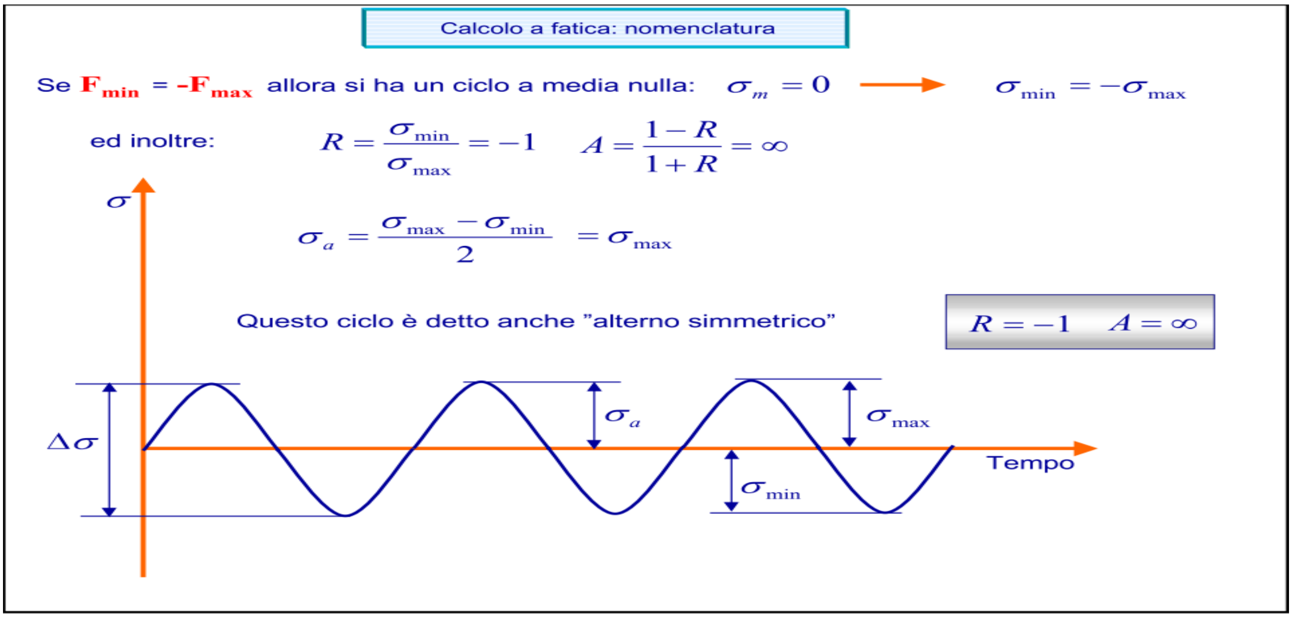

Figure 4.3: Fully reversed stress fatigue

The method consists of adding input information to the black box giving at the output the prevised lifetime of the component and its working duration, without taking into consideration the mechanism that is occurring inside the black box.

A schematic representing the method is shown in Figure 4.4.

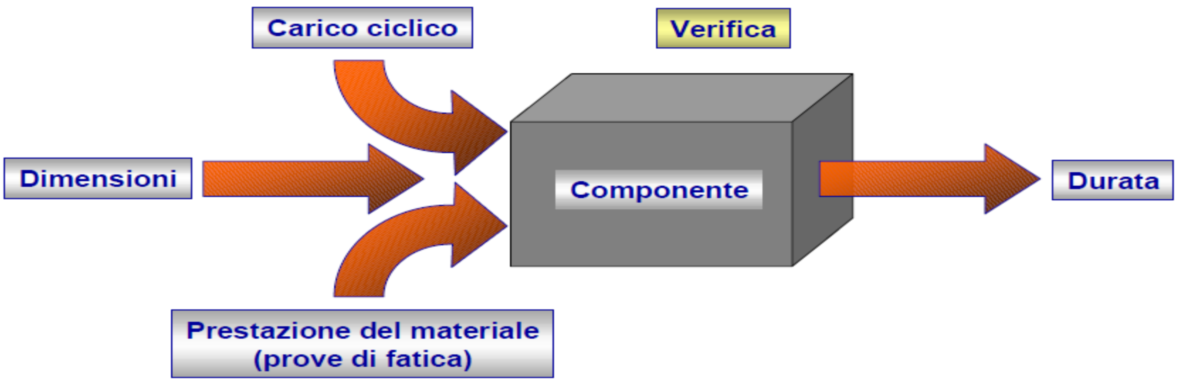

Figure 4.4: Wohler method representation

The design starts with the knowledge of number of cycle N needed for the shaft. After N determination the material type with the proper diameter are chosen in such a way to satisfy application requirement and sustain the period wanted.

For a certain stress variation $\Delta\sigma 1$ a number of cycles N1 is calculated,

For a certain stress variation $\Delta\sigma 2$ a number of cycles N2 is calculated, this variation is represented graphically in Figure 4.5.

Therefore, after the determination of various number of cycles a continuos curve can be plotted like the one shown in Figure 39.The obtained curve is called the **CURVE OF WÖHLER**.[68]

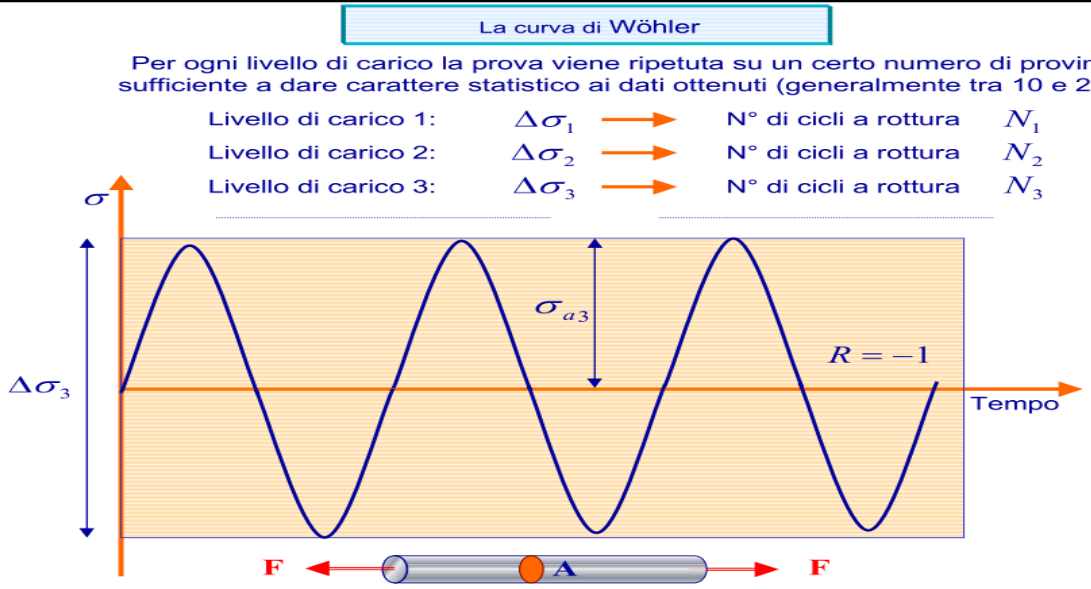

Figure 4.5: Δσ as function of N

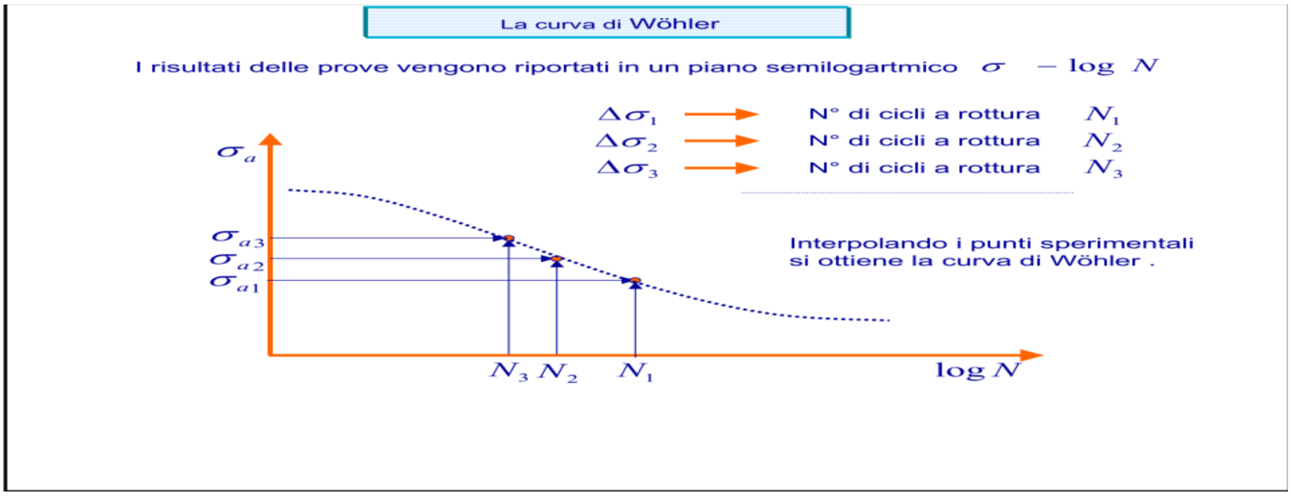

Figure 4.6: Wöhler Curve

Curve of Wöhler is a longevity function. It is intended to represent the number of cycles n that can be made of the different system components before reaching fatigue failure as a function of maximal stress, meanwhile all load cycles present the same amplitude and the same ratio K between extreme values.

Regarding the Curve of Wöhler shown in Figure 4.6 , the following hypothesis are considered:

$$\boldsymbol{For\ n = 8 \times 10^3: \sigma = \sigma R\ and\ \tau = \frac{\sigma R}{\sqrt{3}}}$$

$\boldsymbol{For\ 8 \times 10\text{^}3 < n < 2 \times 10\text{^}6}$, this **zone is called resistance of limited duration** because the stress is close to rupture stress and therefore the element cannot stay a long duration in this state. In this range, a straight line represents the Wohler curve in a reference system made of two logarithmic axes.

The slope of the curve in the considered interval is characterized by c:

➔ in terms of axial stress:

$$c = tan\,\emptyset = (log\,2\times10^6 - log\,8\times10^3)/(log\,\sigma R - log\,\sigma LF)$$

➔ in terms of shear stress:

$$c = \tan\emptyset = (\log 2\times10^6 - \log 8\times10^3)/(\log\frac{\sigma R}{\sqrt{3}} - \log \tau LF)$$

$$For\ n = 2\times10^6: \sigma = \sigma LF\ and\ \tau = \tau LF$$

$For\ n > 2\times10\text{^}6$, called also **zone of resistance with duration**. In this zone the stress load is far from rupture. Therefore a long duration can be achieved having a slope less steep than the limited duration zone.The loads endured are lower. Some references consider this zone of horizontal steepness (slope =0).In this case it is assumed that the number of cycles reached is infinite. And some of them assume it has a small slope which means the number of cycles is not infinite.

In the last case the slope will be

$$c' = \tan\emptyset' = c + \sqrt{}(c^2 + 1)$$

4.5 ELEMENTS SUBJECTED TO FATIGUE

4.5.1 SHAFT FATIGUE ANALYSIS

Calculation results

All Shaft Results with verifications for the case implemented in the Thesis are shown in Table 4.1.[69]

F.E.M. 1.001/87 SHAFT DIMENSIONING			
Symbol	**Caracteristic**	**Unit**	**Value**
Classe		= -- M	7,00
Regime di carico		= --	L4
Condizione di impiego		= --	T5
Durata convenzionale		= h	6300,00
n = Numero giri al minuto		= --	10,00
nt =	n.giri totali dell'albero	= --	3780000,00
Materiale		= --	Fe510
D =	diametro in esame	= mm	400,00
Di =	diametro interno	= mm	0,00
Dm =	diametro maggiore	= mm	405,00
r =	raggio di raccordo	= mm	1,50
D/d =		= --	1,01
q =	tabella A 4131 a	= mm	0,13
r/d equivalente =		= mm	0,13
A =	Area	=mm2	125663,71
W =	Modulo resistenza a flessione	=cm3	6283,19
Wt =	Modulo resistenza a torsione	=cm3	12566,37
a =	Sbalzo	= mm	26,50
Load			
V =	Verticale	= N	287310,00
Stresses			
Mv =	Flessione verticale = gamma*psi*V*a	= N*m	10944,72
Tv =	Taglio verticale = gamma*psi*V	= N	413008,13
gamma =	Coeffic. di maggiorazione	= --	1,25
Breaking admissible tension			
sigR =	tensione di rottura	= MPa	510,00
sigAl =	tensione amm. I e II cc = sigR / 2,2	= MPa	231,82

sigAIII=	tension amm. III cc = sigR / 1,8	= MPa	283,33
TauAI =	SigAI/3^0,5	= MPa	133,84
TauAIII=	SigAIII/3^0,5	= MPa	163,58
Breaking resistance verification			
Combinazione carichi I			
Sig.v =	Mv/w	= MPa	1,74
Tau v =	Tv/A*4/3	= MPa	4,38
SigidI =	Sigv	= MPa	1,74
Fatigue admissible stress			
COEFFICIENTE DI FORMA (Ks)			
Ks =	diagramma A 4131b p.4-27 FEM	= ---	1,60
COEFFICIENTE DI DIMENSIONE (Kd)			
Kd =	tavola T A 4132 p.4-27 FEM	= ---	1,66
COEFFICIENTE DI LAVORAZIONE e CORROSIONE(Ku/Kc)			
Ku =	diagramma A 4132 p 4-28 FEM	= ---	1,10
Sig bw =	tensione limite a fatica = sig R / 2	= MPa	255,00
Sig wk =	Sig bw / (Ks*Kd*Ku*Kc)	= MPa	87,28
k =	Sig min /Sig max	= ---	-1,00
(ck) =	5 /(3-2*k)	= ---	1,00
(ck) =	5/3 /(1-(1-5/3*Sigwk/R)*k)	= ---	0,00
Sig d =	Sig wk * ck	= MPa	87,28
tau d =	tau wk * ck	= MPa	50,39
c =	pendenza curva di Wohler tra 8k e 2M di cicli		
=	tg fi = (lg2*10^6 - lg8000)/(lgsigR-lgSigd)	= ---	3,13
c\ =	pendenza curva di Wohler tra 8k e 2M di cicli		
=	tg fi = (lg2*10^6 - lg8000)/(lgtauR-lgtaud)	= ---	3,13
c' =	pendenza curva di Wohler oltre i 2M di cicli		
	tg fi' = c + (c^2+1)^0.5	= ---	6,41
c'\ =	pendenza curva di Wohler oltre i 2M di cicli		
	tg fi' = c\ + (c\^2+1)^0.5	= ---	6,41
(Kn) =	(2000000 / nt)^(1/c) tra 8k e 2M	= ---	0,00
(Kn') =	(2000000 / nt)^(1/c') oltre 2M	= ---	0,91
(Kn\) =	(2000000 / nt)^(1/c\) tra 8k e 2M	= ---	0,00
(Kn'\) =	(2000000 / nt)^(1/c'\) oltre 2M	= ---	0,91

nu k =	3.2^(1/c) nt =< 2^10^6	= --	0,00
nu k'	3.2^(1/c') nt > 2^10^6	= --	1,20
nu k\	3.2^(1/c\ ) nt =< 2^10^6	= --	0,00
nu k'\	3.2^(1/c'\) nt > 2^10^6	= --	1,20
Sig k =	Sig d*Kn*1	= MPa	79,03
Tau k =	Sig d*Kn*1/3^0.5	= MPa	45,63
Sigaf =	Sigk/ nu k	= MPa	65,92
Tauaf =	Tauk/ nu k\	= MPa	38,06
VERIFICATION RESISTANCE TO FATIGUE			
Sigftot=	Sigv / 1,15	= MPa	1,51
Tau =	Tauv / 1,15	= MPa	3,81
verification Sigma id	< 1,1		0,10

Table 4.1: SHAFT SPREADSHEET

Calculation procedure

a-Determination of Ks (section coefficient)

This coefficient is introduced to take into account the shaft section variation. It is found by the use of certain graphs in which Ks is a function of Metal rupture resistance σR and r/d ratio. A correction factor can be applied in case the D/d ratio changes.

b-Determination of Kd (dimension coefficient)

When the diameter of the shaft increases stress concentration increases for which must be taken into account. This is done by introduction of dimension coefficient .

Tables are used to find this coefficient as function of small diameter d.

c-Determination of Ku (surface shape coefficient)

This coefficient defines the variation of stress concentration as function of surface rigorousness. A graph is plotted in which Ku is found as function of σR.

d-Determination of Kc (corrosion coefficient)

The corrosion can cause serious damage to steel made shafts. This action can be considered by the introduction of Kc coefficient. Graphs are plotted to determine Kc values as function of rupture stress σR.

For rotating shafts the stress is alternated because for a fixed point on shaft the stress type is inverted every half round therefore $R = -1$.

1- σbw:

$$\boldsymbol{\sigma bw = \frac{1}{2} \times \sigma R}$$

2-Proceed to the variable section to determine D/d and r/d

3-Determination of Ks: For the calculated D/d value find the corresponding q value from table afterwards Ks can be determined

4-Determination of Kd: For the given d Kd is determined

5-Determination of Ku: after knowledge of shaft piece type Ku can be determined

6-After the determination of all previous parameters σwk can be calculated from the following formula:

$$\sigma wk = \frac{\sigma bw}{Ks \times Kd \times Ku}$$

In Alternated cycle case $K = -1$:

$$cK = (\frac{5}{3 - 2 \times K})$$

$$\sigma d = cK \times \sigma wk$$

$$\tau d = \sigma d/\sqrt{3}$$

8-calculate c:

$$c = \tan \emptyset = (\log 2 \times 10^6 - \log 8 \times 10^3)\,/(\log \sigma R - \log \sigma LF)$$

9-Then , Critical stress is determined :

Flexion

$$\sigma k = 2^{\frac{8-j}{c}} \times \sigma d$$

where j is determined after knowing the class of mechanism considered which is 7 for the case considered.

Shear

$$\tau k = 2^{\frac{8-j}{c}} \times \tau d = \sigma k/\sqrt{3}$$

10-Afterwards:

$$\nu k = 3.2^{1/c}$$

11- Last step gives the admissible stress the shaft having the current diameter and current section variation can support according to the number of cycles required (depends on class of mechanism chosen for the proper application):[70]

$$\sigma af = \frac{\sigma \mathrm{k}}{\nu k}$$

$$\tau af = \frac{\sigma \mathrm{k}}{\nu' k}$$

$$\sigma f\ total = \frac{\sigma \mathrm{v}}{1.15}$$

$$\tau f\ total = \frac{\tau \mathrm{v}}{1.15}$$

Shaft Fatigue Verifications

$$\sigma f\ total < \sigma\ af$$

$$\tau f\ total < \tau af$$

$$\left(\frac{\sigma f\ total}{\sigma\ af}\right)^2 + \left(\frac{\tau f\ total}{\tau\ af}\right)^2 < 1.1$$

4.5.2 BEARINGS FATIGUE ANALYSIS

Bearing Spreadsheet

Calculation results

All Bearings Results with verifications for the case implemented in the Thesis are shown in Table 4.2.[71]

Table 4.2: BEARING SPREADSHEET

BEARINGS					
Caracteristic	**Symbol**	**Value**	**Unit**	**Note/equation**	
Intern Diameter	d	160	mm		
C	coefficienti dynamico	585	kN		
C0	coefficienti statico	880	kN		
Fr	forza radiale	74,2516	kN		
Fa	forza assiale	0	kN		
alpha	angolo fune storto	0	rad		
Fa/Fr		0	-		
e		0,22	-		
P0	Carico statico equivalente	74,2516	kN		
P	Carico dinamico equivalente	74,2516	kN		
Bearing type		rulli	-		
k		3,3			
Cycle numbers/10^6	L	908,419			
Bearings Verification					
Caracteristic	**Symbol**	**Value**	**Unit**	**Note/equation**	
Velocity	v	20	m/min		
RPM	n	62,6594			
Total number of turns	nt	6015305			
Verification of cycles		0,00662		**1%**	VERIFIED
Static Verification		0,08438		**8%**	VERIFIED
Dynamic Verification		0,12693		**13%**	VERIFIED

Calculation procedure

Point of departure is choosing the Bearing internal diameter d to precise the following parameters:

C0 static coefficient, C dynamic coefficient, e, Y1, Y2, Y0.

1- Radial force of rope on shaft:

$$\mathbf{Fr = (P \times \frac{g}{2} + m\ drum \times \frac{g}{2})}$$

Axial force is calculated knowing twisting angle between shaft and twisted rope alpha:

$$\boldsymbol{Fa = Fr \times \tan\alpha}$$

2- P0 is calculated to verify static stability:

$$P0 = Fr + Y0 \times Fa$$

Static Verification:

$$P0 < C0$$

Dynamic Load :

$$P = Fr + Y1 \times Fa \; if \; e \geq \frac{Fa}{Fr}$$

$$P = 0.67 \times Fr + Y2 \times Fa \; if \; e \leq \frac{Fa}{Fr}$$

Dynamic Verification:

$$P < C$$

The bearings used at the shaft extremity is chosen based on the required number of cycles in millions

The following formula reports the bearing selection based on standards available in Cerrato Company database:

$$L = \left(\frac{c}{p}\right)^{n}$$

Where L is anticipated in million number of cycles during bearing lifetime

C is the dynamic capacity expressed in kN ,it is specific for each bearing number

P is the load applied un kN

n is a constant depending on bearing type :

$$n = 3 \; for \; ball \; bearings, n = 10/3 \; for \; roller \; bearings.$$

The formula can be applied in two reversible direction:

a-If C is known the lifetime of the bearing can be determined.

b-If lifetime is known the bearing having the corresponding C value can be selected.[72]

Fatigue Verification

Bearing Rounds per minute:

$$n = V \times 4/(\pi \times \frac{De}{1000})$$

Total number of Rounds in lifetime:

$$nt = n \times h \times 60$$

With h being, total number of functioning hours during bearing lifetime

Rounds Verification :

$$nt < L$$

4.5.3 JOINT COUPLINGS FATIGUE ANALYSIS

Couplings Spreadsheet

Calculation Results

All Couplings Results with verifications for the case implemented the Thesis are shown in Table 4.3.[73]

Table 4.3: Joint Couplings Spreadsheet

Joint Coupling					
Caracteristic	**Symbol**	**Value**	**Unit**	**Note/equation**	
Size	d	260	mm		
Radial load	F giunto	5500	Kg		
Couple	C giunto	4450	Kg*m		
Joint Verification					
Caracteristic	**Symbol**	**Value**	**Unit**	**Note/equation**	
Nominal Couple	C nom	3363,53	Kg*m		
Radial load	F rad	4418,11	Kg		
Couple Verification		0,75585		**76%**	VERIFIED
Load Verification		0,80329		**80%**	VERIFIED

Calculation procedure

Two main parameters are calculated in the Couplings Design procedure:

1- Radial force exerted on joint:

$$F\ radiale = (F0 \times \frac{nd}{9.81} + m\ drum)/2$$

2- Nominal torque of joint :

$$Cnom = \frac{SF \times Dp \times P \times nt}{2000 \times ng \times T}$$

Out of standards for a selected Joint the following parameters are known:
Diameter, Radial load, and Nominal torque.
To choose the Correct couplings from Standards available in the Company the following conditions must be satisfied:

$$F\ joint > F\ exerted$$
$$C\ joint\ > C\ exerted$$

CONCLUSION

Using the available spreadsheets, the Designer will obtain directly the components dimensions required if similar data is present in CERRATO STANDARDS.

If not specific orders must be made depending on criticality of condition needed to be satisfied. In some cases where safety margins are far from case study some adjustments can be made.

F.E.M 1.001 3rd edition revised 1998.10.01 Normative is used as reference for all components designed. It is used by the Company as the principal Normative used in Crane System design.

Verification are included inside spreadsheet to make sure all data is eligible and to guarantee the design is safe. As said before some exceptions can be made depends on engineer experience and request.

The spreadsheets can be updated with time. Their use is fundamental today because the have easy access and save a lot of time.

CERRATO Company has a list of available standards for each component type. The values for all components are shown in Tables in Chapter 3.

The selected value is the standard value closer to the program value for the case implemented here.

A list of parameters determines the whole design of Crane System. Below is listed the Design Results for each component of the Hoist system aimed to lift 33 tons of industrial mass.

1- Mechanism class: M7
2- Work duration: 6300 hours
3- Load Regime: L4
4- Rope type: S10AR 6*36 WARRINGTON-IWRC Right regular lay Cod54.150
5- Rope diameter: 19 mm.
6- Drum material type: Fe 510 in old standards or S355 in Current standards, S refers for steel material and 355 is the yield strength of steel in MPa.
7- Drum diameters: D min=426 mm, D primitive=453 mm, D inner=407 mm.
8- Shaft diameter: D= 405 mm.
9- Shaft material type: S355
10- Pulley diameter: Dp= 475 mm.
11- Bearing type : Spherical Roller Bearing
12- Bearing diameter: D intern= 160 mm
13- Joint Couplings brand: Maina
14- Joint Couplings diameter: 260 mm
15- Joint Couplings Radial force: 5500 kgF
16- Joint Couplings nominal Torque: 4450 kgF*m

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