Design of Cranes for Industrial Lifting Operations.

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September, 2018

Thesis submitted in compliance with the requirements for the Master of Science degree
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

- **m = 33 t**
- 2 pulls from drum sides
- Coupling
- One drum
- Bearings
- Grooved rope Drum
- Hoist rope
- Pulley
- Bottom block
- Thrust bearing
- 8 rope pulls in total
ACKNOWLEDGMENTS

All source of information were exploited using legal access in the Company’s Technical office including books, Normative, and Catalogs.\textsuperscript{[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.\textsuperscript{[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 shown 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:

$$\sum F = m \times a$$

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:

$$V = V_0 + a \times t$$

V0 is velocity at time t=t0; 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:

$$X = X_0 + V_0 \times t + a/2 \times t^2$$

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

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

$$V = \omega \times R$$

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

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

Translation Power is shown in the formula below:

$$P = F \times V = F \times R \times \omega$$
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.\textsuperscript{[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.\textsuperscript{[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\textsuperscript{[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.\textsuperscript{[16]}
Combination of Elements of the Crane in CERRATO Company is shown in Figure 1.4\cite{17} and Figure 1.5\cite{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

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>Bridge mass Kg</td>
</tr>
<tr>
<td>M2</td>
<td>Cart mass Kg</td>
</tr>
<tr>
<td>M3</td>
<td>Loading mass Kg</td>
</tr>
<tr>
<td>Sc</td>
<td>Beam width m</td>
</tr>
<tr>
<td>D</td>
<td>Wheels diameter m</td>
</tr>
<tr>
<td>M</td>
<td>Class of mechanism used</td>
</tr>
<tr>
<td>Va</td>
<td>Translation velocity m/s</td>
</tr>
<tr>
<td>Ds</td>
<td>Shaft diameter m</td>
</tr>
<tr>
<td>X</td>
<td>Number of motors</td>
</tr>
<tr>
<td>Nrm/Nrt</td>
<td>Ratio of driving wheels over total number of wheels</td>
</tr>
<tr>
<td>C/B</td>
<td>Type of rail on which are wheels mounted</td>
</tr>
<tr>
<td>B/R</td>
<td>Type of wheel connection</td>
</tr>
<tr>
<td>Y/N</td>
<td>External reduction of reducer</td>
</tr>
<tr>
<td>Z1</td>
<td>Number of pinion teeth</td>
</tr>
<tr>
<td>Z2</td>
<td>Number of crown teeth</td>
</tr>
<tr>
<td>Motor Efficiency</td>
<td></td>
</tr>
<tr>
<td>Motor Type</td>
<td></td>
</tr>
<tr>
<td>Mf</td>
<td>Braking torque N*m</td>
</tr>
<tr>
<td>R.R.e</td>
<td>Effective Reduction ratio</td>
</tr>
</tbody>
</table>

#### Table 1.2: SYSTEM OUTPUT DATA

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rope material type and diameter</td>
<td></td>
</tr>
<tr>
<td>Rope stress resistance</td>
<td></td>
</tr>
<tr>
<td>Drum material type and diameter</td>
<td></td>
</tr>
<tr>
<td>Pulleys diameter</td>
<td></td>
</tr>
<tr>
<td>Shaft diameter</td>
<td></td>
</tr>
<tr>
<td>Shaft number of cycles</td>
<td></td>
</tr>
<tr>
<td>Motor and reducer type</td>
<td></td>
</tr>
<tr>
<td>Motor and reducer Power</td>
<td></td>
</tr>
<tr>
<td>Bearings type and dimensions</td>
<td></td>
</tr>
<tr>
<td>Bearings number of cycles</td>
<td></td>
</tr>
<tr>
<td>Couplings type</td>
<td></td>
</tr>
<tr>
<td>Couplings size</td>
<td></td>
</tr>
</tbody>
</table>

#### Table 1.3: WINCH COMPONENTS

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ropes</td>
<td></td>
</tr>
<tr>
<td>Drum</td>
<td></td>
</tr>
<tr>
<td>Rotating Shaft</td>
<td></td>
</tr>
<tr>
<td>Bearings</td>
<td></td>
</tr>
<tr>
<td>Frame &amp; Supports</td>
<td></td>
</tr>
<tr>
<td>Reducer</td>
<td></td>
</tr>
<tr>
<td>Electric Motor control unit</td>
<td></td>
</tr>
<tr>
<td>Gear couplings</td>
<td></td>
</tr>
<tr>
<td>Motor Brake</td>
<td></td>
</tr>
</tbody>
</table>
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]
CHAPTER 2: CRANE COMPONENTS DESIGN

2.1 ROPE DESIGN

Figure 2.1: Steel Rope 3D view

Figure 2.2: Different Rope types

Products :: Steel Wire Ropes

IWRC Galvanized 6 x 19
IWRC Galvanized 6 x 25
IWRC Galvanized 6 x 36
IWRC Galvanized 6 x 41

IWRC Ungalvanized 6 x 19
IWRC Ungalvanized 6 x 25
IWRC Ungalvanized 6 x 36
IWRC Ungalvanized 6 x 41
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. The different rope types are present in Figure 2.2.

The following parameters are checked before choosing the rope:

Breaking strength, minimum breaking strength, and safety factor.

2.1.2 STANDARDS SELECTION

The standards used in Ropes application will be standards of Tecí’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. Tecí’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.

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.

   2.1.5.2 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:

\[ \Delta l = \frac{L \times F}{E \times S} \]

where \( \Delta l \) 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 and 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:

\[ \frac{D}{d} > 25; D/\varnothing > 300 \]

Case of Return Sheaves:

\[ \frac{D}{d} > 20; D/\varnothing > 250 \]

Case of Lifts and Elevators:

\[ \frac{D}{d} > 40; D/\varnothing > 500 \]
Rope Pressure on Sheaves and Winches

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

It is calculated using the equation below:

\[ P = (T_1 + T_2) / (D \times d) \]

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

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

Fleet angle

When the rope is wound 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 exceed 16 always (generally 20 or more).

**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”.

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.

**Diameter Determination**

Two rope selection methods are available depending on constructor choice:

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 $22^\circ$.

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

Definition

$Zp$ is the ratio between the minimal breaking force $F_0$ and maximal traction effort for the cable

$$Zp = \frac{F_0}{S}$$

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.$^{[22]}$

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

Table 2.2: $Zp$ as function of Classe of Mechanism

<table>
<thead>
<tr>
<th>Class of mechanism</th>
<th>Minimal value of Zp</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>3.15</td>
</tr>
<tr>
<td>M2</td>
<td>3.35</td>
</tr>
<tr>
<td>M3</td>
<td>3.55</td>
</tr>
<tr>
<td>M4</td>
<td>4</td>
</tr>
<tr>
<td>M5</td>
<td>4.5</td>
</tr>
<tr>
<td>M6</td>
<td>5.6</td>
</tr>
<tr>
<td>M7</td>
<td>7.1</td>
</tr>
<tr>
<td>M8</td>
<td>9</td>
</tr>
</tbody>
</table>
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

\[ K’ = (\pi \times f \times k)/4 \]

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

\[ C = \sqrt{Zp/(K’ \times r0)} \]

The rope diameter must meet the following condition:

\[ d \geq C \times \sqrt{S} \]

K’ can be extracted from the ISO 2408 recommendation.

Minimum Winding diameter

It is determined using the following relation:

\[ D \geq H \times d \]

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

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

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]

\[ C_t = \left( \frac{K \times T}{P \times t} \right) \]

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 its diameter and its 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:

\[ T = \frac{D}{2} \times F \]

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 guarantee 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

<table>
<thead>
<tr>
<th>Class of mechanism</th>
<th>Drum Hd</th>
<th>Returning Pulley</th>
<th>Equilibration Pulley</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>11.2</td>
<td>12.5</td>
<td>11.2</td>
</tr>
<tr>
<td>M2</td>
<td>12.5</td>
<td>14</td>
<td>12.5</td>
</tr>
<tr>
<td>M3</td>
<td>14</td>
<td>16</td>
<td>12.5</td>
</tr>
<tr>
<td>M4</td>
<td>16</td>
<td>18</td>
<td>14</td>
</tr>
<tr>
<td>M5</td>
<td>18</td>
<td>20</td>
<td>14</td>
</tr>
<tr>
<td>M6</td>
<td>20</td>
<td>22.4</td>
<td>16</td>
</tr>
<tr>
<td>M7</td>
<td>22.4</td>
<td>25</td>
<td>16</td>
</tr>
<tr>
<td>M8</td>
<td>25</td>
<td>28</td>
<td>18</td>
</tr>
</tbody>
</table>

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

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 $H_p$ 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:

$$D_p = H_p \times d$$
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]
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 the shaft is subjected to both torsion and flexion (not negligible). In this case the dimensioning is done using flexion-torsion criteria.\textsuperscript{[44]}

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

\begin{table}[h]
\centering
\begin{tabular}{|c|c|c|c|}
\hline
Sollecitazione & Progetto & Verifica & Significato dei simboli \\
\hline
\hline
1 - Flessione & \( d = \sqrt{\frac{10M_I}{\sigma_{am}}} \) & \( \sigma = \frac{M_I}{W_t} \leq \sigma_{am} \) & \( d = \) diametro [mm] \\
& \( \sigma_{am} = \frac{R}{\eta} \) & & \( M_I = \) momento flettente massimo [Nmm] \\
& & & \( W_t = \) modulo di resistenza alla flessione [mm\(^3\)] \\
& & & \( \sigma_{am} = \) carico unitario di sicurezza normale ammissibile [N/mm\(^2\)] \\
& & & \( R = \) carico di rottura [N/mm\(^2\)] \\
& & & \( \eta = \) grado di sicurezza: \\
& & & \hspace{1cm} 3 \div 5 \text{ per carichi statici} \\
& & & \hspace{1cm} 9 \div 15 \text{ per carichi dinamici} \\
\hline
2 - Torsione & \( d = \sqrt{\frac{5M_t}{\tau_{am}}} \) & \( \tau = \frac{M_t}{W_t} \leq \tau_{am} \) & \( d = \) diametro teorico [mm] \\
& \( \tau_{am} = 4/5 \cdot \sigma_{am} \) & & \( M_t = \) momento torcente [Nmm] \\
& & & \( W_t = \) modulo di resistenza alla torsione [Nmm] \\
& & & \( \tau_{am} = \) carico unitario di sicurezza tangenziale ammissibile [N/mm\(^2\)] \\
& & & \( P = \) potenza trasmessa [kW] \\
& & & \( n = \) velocità di rotazione [giri/\text{min}] \\
\hline
3 - Flesso-Torsione & \( d = \sqrt{\frac{10M_{fid}}{\sigma_{l}}} \) & \( \sigma = \sqrt{\sigma^2 + \tau^2} \leq \sigma_{am} \) & \( M_{fid} = \) momento flettente ideale: \hspace{1cm} \text{il momento flettente che da solo crea sull'albero lo stesso stato di pericolosità della contemporanea presenza dei momenti torcente e flettente} \\
& & & \( \sigma = \frac{M_t}{W_t} \) \\
& \( M_{fid} = \sqrt{M_t^2 + 0,75M_t^2} \) & \( \tau = \frac{M_t}{W_t} \) & \( \theta = \frac{M_t \cdot l/G \cdot J_p}{4 \cdot 180} \leq \sigma_{am} \) \\
& & & \( I = [\text{mm}]; \ G = [\text{N/mm}^2]; \ J_p = [\text{mm}^4] \) \\
\hline
\end{tabular}
\caption{Equations determining bending & twisting moments}
\end{table}

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

\textbf{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.

![Figure 2.10: Shaft Section variation](image)

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:

\[
\frac{(D - d)}{(d \times 100)} \leq 20
\]

### 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 its 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:

\[ l > F/(P \times d) \]

where \( F \) force applied, \( P \) am admissible pressure and \( d \) pin diameter.\[47]\]

\[ Ps \leq \frac{F}{d \times l} \leq P \]

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:

\[ Ce = C + \frac{Mn}{6} + \frac{Cr + Mo + V}{5} + \frac{Ni + Cu}{15} \]

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

Weldability Verification:

\[ If Ce < 0.4 \text{ Optimal Welding} \]

\[ If Ce > 0.4 \]

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

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

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:

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 varied 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:

\[ M = T \times D/2 \]

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:

\[ P = M \times \omega \]

M torque in N*m, \( \omega \) 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

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

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](image)

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

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.\textsuperscript{[62]}

A Gear coupling example is shown in Figure 2.21.\textsuperscript{[63]}

\begin{figure}[h]
\centering
\includegraphics[width=0.8\textwidth]{gear_coupling.png}
\caption{Gear Couplings}
\end{figure}
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 Section 2.\[64\]

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

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load</td>
<td>$P$</td>
<td>33000</td>
<td>kg</td>
<td></td>
</tr>
<tr>
<td>width</td>
<td>$S$</td>
<td>6</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Lift velocity</td>
<td>$V_{\text{sol}}$</td>
<td>3,5</td>
<td>m/min</td>
<td></td>
</tr>
<tr>
<td>Trolley translational velocity</td>
<td>$V_{t\text{-carrello}}$</td>
<td>20</td>
<td>m/min</td>
<td></td>
</tr>
<tr>
<td>Bridge translational velocity</td>
<td>$V_{t\text{-ponte}}$</td>
<td>40</td>
<td>m/min</td>
<td></td>
</tr>
<tr>
<td>Hook range</td>
<td>$c$</td>
<td>6</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>Voltage of alimentation</td>
<td></td>
<td>400</td>
<td>V</td>
<td></td>
</tr>
<tr>
<td>Frequency of alimentation</td>
<td></td>
<td>50</td>
<td>Hz</td>
<td></td>
</tr>
<tr>
<td>Structural Class</td>
<td></td>
<td>A 7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Class of Mechanism</td>
<td></td>
<td>M 7</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.2: TECHNICAL DATA INPUT

<table>
<thead>
<tr>
<th>INPUT PARAMETERS ROPE-DRUM</th>
<th>SYMBOL</th>
<th>VALUE</th>
<th>Unit</th>
<th>MINIMAL VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rope diameter</td>
<td>$d$</td>
<td>19</td>
<td>mm</td>
<td>14,6</td>
</tr>
<tr>
<td>Minimal breakage resistance</td>
<td>$R_0$</td>
<td>2150</td>
<td>MPa</td>
<td></td>
</tr>
<tr>
<td>Filling factor</td>
<td>$f$</td>
<td>0,59</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Minimum calculated break stress</td>
<td>$\gamma$</td>
<td>0,8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$N^#$ drums/$N^#$ pull per each drum/$N^#$ pull at hook</td>
<td>combo</td>
<td>1x2x8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$N^#$ principal drums</td>
<td>$N_{\text{princ-tamb}}$</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>External diameter for raw tube</td>
<td>$D_e$</td>
<td>457,2</td>
<td>mm</td>
<td>216</td>
</tr>
<tr>
<td>Raw tube thickness</td>
<td>$s_{\text{tubo}}$</td>
<td>25</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Distance between central ropes (0 per 1 tiro dal tamburo)</td>
<td>$x$</td>
<td>200</td>
<td>mm</td>
<td>168</td>
</tr>
<tr>
<td>Distance between real supports</td>
<td>$i$</td>
<td>2200</td>
<td>mm</td>
<td>659</td>
</tr>
<tr>
<td>Shaft length till bearing center</td>
<td>$a$</td>
<td>500</td>
<td>mm</td>
<td>259</td>
</tr>
<tr>
<td>Distance drum axis/block pulley</td>
<td>$b$</td>
<td>1500</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Distance drum axis/fixed pulley</td>
<td>$c$</td>
<td>5000</td>
<td>mm</td>
<td></td>
</tr>
</tbody>
</table>
### Table 3.3: ROPE DATA

<table>
<thead>
<tr>
<th>VERIFICATIONS ROPE</th>
<th>SYMBOL</th>
<th>VALUE</th>
<th>Unit</th>
<th>MINIMAL VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Class of mechanisms</td>
<td>M</td>
<td>7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Load to hook (add suspended masses)</td>
<td>P</td>
<td>33000</td>
<td>kg</td>
<td></td>
</tr>
<tr>
<td>No. of drums</td>
<td>T</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>N ° shots from each drum</td>
<td>N_{tri-tamb}</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>No. of hooks</td>
<td>N_{tri-gancio}</td>
<td>8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coefficient ZP prospect 1 ISO 4308</td>
<td>zp</td>
<td>7.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pulling force on the rope</td>
<td>F_0</td>
<td>40466</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>Real tensile load on the rope to be chosen in the catalog</td>
<td>F_{reale}</td>
<td>287310</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>Minimum rope diameter</td>
<td>d_0</td>
<td>19</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Cable diameter check</td>
<td>d&gt;d_0</td>
<td>VERIFICATO</td>
<td>107%</td>
<td></td>
</tr>
<tr>
<td>CHARACTERISTICS</td>
<td>SYMBOL</td>
<td>VALUE</td>
<td>Unit</td>
<td>MINIMAL VALUE</td>
</tr>
<tr>
<td>-----------------------------------------------------</td>
<td>--------</td>
<td>--------</td>
<td>------------</td>
<td>---------------</td>
</tr>
<tr>
<td>Class of mechanisms</td>
<td>M</td>
<td>7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hook travel</td>
<td>c</td>
<td>6000</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Drum material</td>
<td>mat</td>
<td>S355JR (Fe510B)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lifting speed</td>
<td>V_soll</td>
<td>3,5</td>
<td>m/min</td>
<td></td>
</tr>
<tr>
<td>mm processing raw tube on the diameter</td>
<td>lav</td>
<td>6</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Tool diameter for machining</td>
<td>2xr_g</td>
<td>20,4</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Coefficient h1 table 2 ISO 4308</td>
<td>h_1</td>
<td>22</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coefficient h2 prospect 2 ISO 4308</td>
<td>h_2</td>
<td>25</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coefficient h3 table 2 ISO 4308</td>
<td>h_3</td>
<td>16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rope pitch</td>
<td>p</td>
<td>21,5</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Minimum drum diameter</td>
<td>D_1</td>
<td>426</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Minimum diameter for transmission pulleys</td>
<td>D_2</td>
<td>475</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Minimum diameter for balancing pulleys</td>
<td>D_3</td>
<td>304</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Diameter ridges</td>
<td>D_c</td>
<td>451</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Primitive theoretical diameter</td>
<td>D_p</td>
<td>453</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Primitive diameter rounded down</td>
<td>D_i</td>
<td>407,2</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Throat bottom diameter</td>
<td>D_g</td>
<td>434</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Thickness of tube under the ring</td>
<td>s</td>
<td>13</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Single drum section resistance module</td>
<td>W_g</td>
<td>1806148</td>
<td>mm³</td>
<td></td>
</tr>
<tr>
<td>Number of rope sections for each single rope pull</td>
<td>X</td>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>from the drum</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Indicative media distance single drum</td>
<td>L</td>
<td>1425</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Horizontal excursion of a rope on the single drum</td>
<td>L_1</td>
<td>363</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>N ° of drum turns needed a rope on the single drum</td>
<td>N_s</td>
<td>16,86</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Indicative drum length without shaft</td>
<td>L_tamb</td>
<td>1700</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>Deflection angle movable pulley (block)</td>
<td>α_b</td>
<td>2,77</td>
<td>gradi</td>
<td></td>
</tr>
<tr>
<td>Fixed pulley deviation angle</td>
<td>α_f</td>
<td>2,08</td>
<td>gradi</td>
<td></td>
</tr>
<tr>
<td>Maximum deviation angle</td>
<td>α_amm</td>
<td>3,00</td>
<td>gradi</td>
<td></td>
</tr>
<tr>
<td>Check angle deflection mobile pulley (block)</td>
<td>α_b&lt;α_amm</td>
<td>VERIFICATO</td>
<td>62%</td>
<td></td>
</tr>
<tr>
<td>Check angle of fixed pulley deviation</td>
<td>α_f&lt;α_amm</td>
<td>VERIFICATO</td>
<td>46%</td>
<td></td>
</tr>
<tr>
<td>Indicative drum weight</td>
<td>m</td>
<td>586</td>
<td>kg</td>
<td></td>
</tr>
<tr>
<td>Number of drum revolutions</td>
<td>n</td>
<td>9,84</td>
<td>giri/min</td>
<td></td>
</tr>
<tr>
<td>Operating hours according to class M</td>
<td>t</td>
<td>6300</td>
<td>h</td>
<td></td>
</tr>
<tr>
<td>Total number of cycles</td>
<td>N</td>
<td>3718534</td>
<td>cicli</td>
<td></td>
</tr>
<tr>
<td>Static admissible tension</td>
<td>σ_{amm-st}</td>
<td>240</td>
<td>MPa</td>
<td></td>
</tr>
<tr>
<td>Permissible tension fatigue 2x10⁶</td>
<td>σ_{2x10^6}</td>
<td>160</td>
<td>MPa</td>
<td></td>
</tr>
<tr>
<td>Fatigue allowable stress 5x10⁶</td>
<td>σ_{5x10^6}</td>
<td>118</td>
<td>MPa</td>
<td></td>
</tr>
</tbody>
</table>
### Table 3.4: DRUM DATA

<table>
<thead>
<tr>
<th>Description</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Admissible tension fatigue 1x10^8</td>
<td>$\sigma_d$</td>
<td>65</td>
<td>MPa</td>
</tr>
<tr>
<td>Admissible stress fatigue</td>
<td>$\sigma_d$</td>
<td>130</td>
<td>MPa</td>
</tr>
<tr>
<td>Minimum allowable tension</td>
<td>$\sigma_{amm}$</td>
<td>130</td>
<td>MPa</td>
</tr>
<tr>
<td>Local compression tension A</td>
<td>$\sigma_{cA}$</td>
<td>70.2</td>
<td>MPa</td>
</tr>
<tr>
<td>Local compression tension B</td>
<td>$\sigma_{cB}$</td>
<td>119.4</td>
<td>MPa</td>
</tr>
<tr>
<td>Local bending tension A-A</td>
<td>$\sigma_{f1}$</td>
<td>38</td>
<td>MPa</td>
</tr>
<tr>
<td>Global bending tension A-A</td>
<td>$\sigma_{fg}$</td>
<td>22.4</td>
<td>MPa</td>
</tr>
<tr>
<td>Global ideal tension A</td>
<td>$\sigma_{ida}$</td>
<td>95.7</td>
<td>MPa</td>
</tr>
<tr>
<td>Local ideal tension B</td>
<td>$\sigma_{idB}$</td>
<td>113.3</td>
<td>MPa</td>
</tr>
<tr>
<td>Verification of ideal global static tension A</td>
<td>$\sigma_{ida} &lt; \sigma_{amm}$</td>
<td>VERIFICATO</td>
<td>49%</td>
</tr>
<tr>
<td>Verification of ideal static local tension test B</td>
<td>$\sigma_{idB} &lt; \sigma_{amm}$</td>
<td>VERIFICATO</td>
<td>64%</td>
</tr>
<tr>
<td>Verification of the ideal ideal global tension A</td>
<td>A</td>
<td>VERIFICATO</td>
<td>17%</td>
</tr>
<tr>
<td>Check the ideal local fatigue tension B</td>
<td>B</td>
<td>VERIFICATO</td>
<td>41%</td>
</tr>
<tr>
<td>Safety factor of the drum joint</td>
<td>SF</td>
<td>1.8</td>
<td></td>
</tr>
<tr>
<td>Nominal torque to the drum joint</td>
<td>C_{nom}</td>
<td>3364</td>
<td>kgm</td>
</tr>
<tr>
<td>Radial load to the drum joint</td>
<td>F_{radiale}</td>
<td>4418</td>
<td>kg</td>
</tr>
<tr>
<td>Joint Mania size</td>
<td>Mania</td>
<td>260</td>
<td></td>
</tr>
</tbody>
</table>
3.2 CALCULATION PROCEDURE

3.2.1 ROPE DATA CALCULATIONS

1- Tension force on one rope is determined:
   \[ F_0 = \frac{(P \times g)}{Ng} \]

2- Real force acting on rope Calculation:
   \[ F_r = F_0 \times Zp \]

3- Minimum Rope diameter:
   \[ d_{min} = \sqrt{\frac{(4 \times Zp \times F_0)}{\sqrt{\pi \times f \times \gamma}}} \]

4- Verification of Rope diameter:
   \[ d > d_0 \]

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:
    \[ Wg = \pi \times (Dg^4 - Dl^4) / (32 \times Dg) \]

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:
    \[ ab = \tan^{-1} \left( L1 / (c + b) \right) \]
16- Fixed Pulley deviation angle: \[ af = \tan^{-1}\left(\frac{L1}{2c}\right) \]
17- Mobile Pulley angle Verification: \[ ab < \alpha_{adm} \]
18- Fixed Pulley angle Verification: \[ af < \alpha_{adm} \]
19- Drum weight: 
   \[ m = \left(\frac{De}{2}\right)^2 - \left(\frac{Di}{2}\right)^2 \times \left(\pi \times \frac{i}{7.85 \times 10^6}\right) \]
20- Number of drum revolutions: 
   \[ n = Vsol \times 10^3 \times Ng/(T \times Nt \times Dp \times \pi) \]
21- Local compression tension at point A: \[ \sigma_cA = 1/2 \times Fo/(P \times S) \]
22- Local compression tension at point B: \[ \sigma_cB = 0.85 \times Fo/(P \times S) \]
23- Local bending tension at section A-A: \[ \sigma_f1 = 0.96 \times Fo/\sqrt{(s^3 \times Dg)} \]
24- Global bending tension at section A-A: \[ \sigma_fg = F0 \times (i - x)/(2 \times Wg) \]
25- Global ideal tension at point A: 
   \[ \sigma_{idA} = \sqrt{\left(\sigma_cA^2 + (\sigma_f1 + \sigma_fg)^2 + \sigma_cA \times (\sigma_f1 + \sigma_fg)\right)} \]
26- Local ideal tension at point B: \[ \sigma_{idB} = \sigma_cB \]
27- Verification of ideal global static tension at point A: \[ \sigma_{idA} < \sigma_{adm} \]
28- Verification of ideal local static tension at point B: \[ \sigma_{idB} < \sigma_{adm} \]
29- Nominal torque exerted on joint: 
   \[ C_{nom} = (SF \times Dp \times P \times Nt)/(2000 \times Ng \times T) \]
30- Radial force exerted on joint: \[ F_{radiale} = 1/2 \times \left(F0 \times \frac{Nt}{g} + m\right) \]
3.2.3 SHAFT DATA CALCULATIONS

1- Total number of shaft rotations:
   \[ nt = n \times h \times 60 \]

2- Diameter variation along sections:
   \[ \frac{D}{d} = \frac{Dm}{D} \]

3- \( r/d \) equivalent:
   \[ \frac{r}{d} = \left( \frac{Dm}{D} \right) + q \]

4- Area:
   \[ A = \pi \times \left( \left( \frac{D}{2} \right)^2 - \left( \frac{Di}{2} \right)^2 \right) \]

5- Flexional Resistance Modulus:
   \[ W = \pi \times 32 \times \left( D^4 - Dt^4 \right) / (D \times 1000) \]

6- Torsional Resistance Modulus:
   \[ Wt = 2 \times W \]

3.2.4 BENDING MOMENT & SHEAR FORCE CALCULATIONS

1- Vertical flexion:
   \[ Mv = a \times \frac{V}{1000} \times 1.15 \times \gamma \]

2- Vertical shear:
   \[ Tv = V \times 1.15 \times \gamma \]

3.2.5 ADMISSIBLE STRESSES CALCULATIONS

1- First admissible tensional stress:
   \[ \sigma A1 = \frac{\sigma R}{2.2} \]

2- Third admissible tensional stress:
   \[ \sigma A3 = \frac{\sigma R}{1.8} \]

3- First admissible shear stress:
   \[ \tau A1 = \sigma A1 / \sqrt{3} \]

4- Third admissible shear stress:
   \[ : \tau A3 = \sigma A3 / \sqrt{3} \]

3.2.6 BREAKAGE RESISTANCE CALCULATIONS

1- Tensional stress:
   \[ \sigma A1 = \frac{Mv}{W} \]

2- Shear stress:
   \[ \tau v = \frac{Tv}{A} \times \frac{4}{3} \]

3- Ideal stress:
   \[ \sigma idl = \sigma v \]
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](image-url)
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, its surface shape, its 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 $F_{\text{max}}, F_{\text{min}}, \sigma_{\text{max}}, \sigma_{\text{min}}, R, A$ are introduced in the Figure 4.1.\(^67\)
4.2 GENERAL CASE

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.

$F_{\text{min}}$ is the minimal force on element while $F_{\text{max}}$ is the maximal force, $F_{\text{min}}$ gives $\sigma_{\text{min}}$ and $F_{\text{max}}$ gives $\sigma_{\text{max}}$

$\sigma_a$ is called alternated stress, $\sigma_m$ is the median stress,

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

$\Delta \sigma$ 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

Figure 4.1: General curve
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.

\[
\begin{align*}
\text{Se } F_{\text{min}} &= 0 \text{ allora si ha: } & \sigma_{\text{min}} &= 0 & \sigma_m &= \sigma_a \\
& & \sigma_{\text{max}} &= \Delta\sigma & \sigma_a &= \frac{\sigma_{\text{max}}}{2} \\
& & R &= \frac{\sigma_{\text{min}}}{\sigma_{\text{max}}} = 0 & A &= \frac{\sigma_a}{\sigma_m} = \frac{1 - R}{1 + R} = 1
\end{align*}
\]

![Diagram of unidirectional stress fatigue with equations and parameters](image)

**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

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.

For a certain stress variation $\Delta \sigma_1$ a number of cycles $N_1$ is calculated,

For a certain stress variation $\Delta \sigma_2$ a number of cycles $N_2$ 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: $\Delta \sigma$ 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:

\[
For \; n = 8 \times 10^3: \sigma = \sigma_R \; and \; \tau = \frac{\sigma R}{\sqrt{3}}
\]

For $8 \times 10^3 < n < 2 \times 10^6$, 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 $c$:  

\[
\text{The slope of the curve in the considered interval is characterized by } c:
\]
in terms of axial stress:
\[ c = \tan \phi = \frac{\log 2 \times 10^6 - \log 8 \times 10^3}{\log \sigma R - \log \sigma LF} \]

in terms of shear stress:
\[ c = \tan \phi = \frac{\log 2 \times 10^6 - \log 8 \times 10^3}{\log \frac{\sigma R}{\sqrt{3}} - \log \tau LF} \]

For \( n = 2 \times 10^6 \): \( \sigma = \sigma LF \) and \( \tau = \tau LF \)

For \( n > 2 \times 10^6 \), 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
\[ c' = \tan \phi' = c + \sqrt{c^2 + 1} \]
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]

<table>
<thead>
<tr>
<th>F.E.M. 1.001/87 SHAFT DIMENSIONING</th>
</tr>
</thead>
<tbody>
<tr>
<td>Symbol</td>
</tr>
<tr>
<td>Classe</td>
</tr>
<tr>
<td>Regime di carico</td>
</tr>
<tr>
<td>Condizione di impiego</td>
</tr>
<tr>
<td>Durata convenzionale</td>
</tr>
<tr>
<td>n</td>
</tr>
<tr>
<td>nt</td>
</tr>
<tr>
<td>Materiale</td>
</tr>
<tr>
<td>D</td>
</tr>
<tr>
<td>Di</td>
</tr>
<tr>
<td>Dm</td>
</tr>
<tr>
<td>r</td>
</tr>
<tr>
<td>D/d</td>
</tr>
<tr>
<td>q</td>
</tr>
<tr>
<td>r/d equivalente</td>
</tr>
<tr>
<td>A</td>
</tr>
<tr>
<td>W</td>
</tr>
<tr>
<td>Wt</td>
</tr>
<tr>
<td>a</td>
</tr>
<tr>
<td>Load</td>
</tr>
<tr>
<td>Stresses</td>
</tr>
<tr>
<td>Mv</td>
</tr>
<tr>
<td>Tv</td>
</tr>
<tr>
<td>gamma</td>
</tr>
<tr>
<td>Breaking admissible tension</td>
</tr>
<tr>
<td>sigR</td>
</tr>
<tr>
<td>sigAI</td>
</tr>
<tr>
<td>sigAII</td>
</tr>
<tr>
<td>-------</td>
</tr>
<tr>
<td>TauAI</td>
</tr>
<tr>
<td>TauAII</td>
</tr>
</tbody>
</table>

### Breaking resistance verification

Combination of loads I

<table>
<thead>
<tr>
<th>Sig.v</th>
<th>Mv/w = MPa</th>
<th>1,74</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tau v</td>
<td>Tv/A*4/3 = MPa</td>
<td>4,38</td>
</tr>
<tr>
<td>SigidI</td>
<td>Sigv = MPa</td>
<td>1,74</td>
</tr>
</tbody>
</table>

### 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 |

<table>
<thead>
<tr>
<th>(ck)</th>
<th>5 / (3-2*k) = ---</th>
<th>1,00</th>
</tr>
</thead>
<tbody>
<tr>
<td>(ck)</td>
<td>5/3 / (1-1-5/3*Sigwk/R)*k = ---</td>
<td>0,00</td>
</tr>
<tr>
<td>Sig d</td>
<td>Sig wk * ck = MPa</td>
<td>87,28</td>
</tr>
<tr>
<td>tau d</td>
<td>tau wk * ck = MPa</td>
<td>50,39</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>c</th>
<th>pendenza curva di Wohler tra 8k e 2M di cicli</th>
</tr>
</thead>
<tbody>
<tr>
<td>=</td>
<td>tg fi = (lg2*10^6 - lg8000)/(lgSigR-lgSigd) = ---</td>
</tr>
<tr>
<td>c\</td>
<td>pendenza curva di Wohler tra 8k e 2M di cicli</td>
</tr>
<tr>
<td>=</td>
<td>tg fi = (lg2*10^6 - lg8000)/(lgtauR-lgtaud) = ---</td>
</tr>
<tr>
<td>c'</td>
<td>pendenza curva di Wohler oltre i 2M di cicli</td>
</tr>
<tr>
<td>=</td>
<td>tg fi' = c + (c^2+1)^0.5 = ---</td>
</tr>
<tr>
<td>c'</td>
<td></td>
</tr>
<tr>
<td>=</td>
<td>tg fi' = c\ + (c^2+1)^0.5 = ---</td>
</tr>
<tr>
<td>(Kn)</td>
<td>(2000000 / nt)^{1/c} tra 8k e 2M = ---</td>
</tr>
<tr>
<td>(Kn')</td>
<td>(2000000 / nt)^{(1/c')} oltre 2M = ---</td>
</tr>
<tr>
<td>(Kn)</td>
<td>(2000000 / nt)^{(1/c')} tra 8k e 2M = ---</td>
</tr>
<tr>
<td>(Kn')</td>
<td>(2000000 / nt)^{(1/c')'} oltre 2M = ---</td>
</tr>
<tr>
<td></td>
<td>Formula</td>
</tr>
<tr>
<td>----------------</td>
<td>-------------------------------------------------------------------------</td>
</tr>
<tr>
<td>( \nu_k )</td>
<td>( 3.2^{(1/c)} ) ( n_t \leq 2 \times 10^6 )</td>
</tr>
<tr>
<td>( \nu_k' )</td>
<td>( 3.2^{(1/c')} ) ( n_t &gt; 2 \times 10^6 )</td>
</tr>
<tr>
<td>( \nu_k' )</td>
<td>( 3.2^{(1/c')} ) ( n_t \leq 2 \times 10^6 )</td>
</tr>
<tr>
<td>( \nu_k' )</td>
<td>( 3.2^{(1/c')} ) ( n_t &gt; 2 \times 10^6 )</td>
</tr>
<tr>
<td>( \sigma_k )</td>
<td>( \sigma_d ) * ( K_n ) * 1</td>
</tr>
<tr>
<td>( \tau_k )</td>
<td>( \sigma_d ) * ( K_n ) * ( 1/3 )^{0.5}</td>
</tr>
<tr>
<td>( \sigma_{af} )</td>
<td>( \sigma_k / \nu_k )</td>
</tr>
<tr>
<td>( \tau_{af} )</td>
<td>( \tau_k / \nu_k )</td>
</tr>
</tbody>
</table>

**VERIFICATION RESISTANCE TO FATIGUE**

<table>
<thead>
<tr>
<th></th>
<th>Formula</th>
<th>Condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \sigma_{ftot} )</td>
<td>( \sigma_v / 1.15 )</td>
<td></td>
<td>1.51</td>
</tr>
<tr>
<td>( \tau )</td>
<td>( \tau_v / 1.15 )</td>
<td></td>
<td>3.81</td>
</tr>
</tbody>
</table>

 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 \( \sigma_R \) 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 \( \sigma_R \).

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 \( \sigma_R \).

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- \( \sigma_{bw} \):

\[
\sigma_{bw} = \frac{1}{2} \times \sigma_R
\]

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 \( \sigma_{wk} \) can be calculated from the following formula:
In Alternated cycle case $K = -1$:

\[
cK = \left(\frac{5}{3 - 2 \times K}\right)
\]

\[
\sigma d = cK \times \sigma wk
\]

\[
\tau d = \sigma d / \sqrt{3}
\]

8-calculate c:

\[
c = \tan \theta = (\log 2 \times 10^6 - \log 8 \times 10^3) / (\log \sigma R - \log \sigma LF)
\]

9-Then, Critical stress is determined:

**Flexion**

\[
\sigma k = 2^8 \times c \times \sigma d
\]

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

**Shear**

\[
\tau k = 2^8 \times \tau d = \sigma k / \sqrt{3}
\]

10-Afterwards:

\[
v k = 3.2^{1/c}
\]

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]

\[
\sigma af = \frac{\sigma k}{v k}
\]

\[
\tau af = \frac{\sigma k}{v' k}
\]

\[
\sigma f total = \frac{\sigma v}{1.15}
\]

\[
\tau f total = \frac{\tau v}{1.15}
\]

**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

<table>
<thead>
<tr>
<th>Caracteristic</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
<th>Note/equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intern Diameter</td>
<td>d</td>
<td>160</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>coefficienti dynamico</td>
<td>585</td>
<td>kN</td>
<td></td>
</tr>
<tr>
<td>C0</td>
<td>coefficienti statico</td>
<td>880</td>
<td>kN</td>
<td></td>
</tr>
<tr>
<td>Fr</td>
<td>forza radiale</td>
<td>74,2516</td>
<td>kN</td>
<td></td>
</tr>
<tr>
<td>Fa</td>
<td>forza assiale</td>
<td>0</td>
<td>kN</td>
<td></td>
</tr>
<tr>
<td>alpha</td>
<td>angolo fune storto</td>
<td>0</td>
<td>rad</td>
<td></td>
</tr>
<tr>
<td>Fa/Fr</td>
<td></td>
<td>0</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>e</td>
<td></td>
<td>0,22</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>P0</td>
<td>Carico statico equivalente</td>
<td>74,2516</td>
<td>kN</td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>Carico dinamico equivalente</td>
<td>74,2516</td>
<td>kN</td>
<td></td>
</tr>
<tr>
<td>Bearing type</td>
<td></td>
<td>rulli</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>k</td>
<td></td>
<td>3,3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cycle numbers/10^6</td>
<td>L</td>
<td>908,419</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

#### Bearings Verification

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
<th>Note/equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity</td>
<td>v</td>
<td>20</td>
<td>m/min</td>
<td></td>
</tr>
<tr>
<td>RPM</td>
<td>n</td>
<td>62,6594</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total number of turns</td>
<td>nt</td>
<td>6015305</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Verification of cycles</td>
<td></td>
<td>0,00662</td>
<td>1%</td>
<td>VERIFIED</td>
</tr>
<tr>
<td>Static Verification</td>
<td></td>
<td>0,08438</td>
<td>8%</td>
<td>VERIFIED</td>
</tr>
<tr>
<td>Dynamic Verification</td>
<td></td>
<td>0,12693</td>
<td>13%</td>
<td>VERIFIED</td>
</tr>
</tbody>
</table>

**Calculation procedure**

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

- \( C_0 \) static coefficient, \( C \) dynamic coefficient, \( e, Y_1, Y_2, Y_0 \).

1. Radial force of rope on shaft:

\[
Fr = \left( \frac{P}{2} + m \text{ drum} \times \frac{g}{2} \right)
\]

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

\[
Fa = Fr \times \tan \alpha
\]

2. \( P_0 \) is calculated to verify static stability:
\[ P_0 = Fr + Y_0 \times Fa \]

**Static Verification:**

\[ P_0 < C_0 \]

**Dynamic Load:**

\[ P = Fr + Y_1 \times Fa \text{ if } e \geq \frac{Fa}{Fr} \]

\[ P = 0.67 \times Fr + Y_2 \times Fa \text{ if } e \leq \frac{Fa}{Fr} \]

**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 in kN.

n is a constant depending on bearing type:

\[ n = 3 \text{ for ball bearings, } n = 10/3 \text{ for roller bearings}. \]

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:

\[ n = \frac{V \times 4}{(\pi \times \frac{De}{1000})} \]

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

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Size</td>
<td>d</td>
<td>260</td>
<td>mm</td>
</tr>
<tr>
<td>Radial load</td>
<td>F giunto</td>
<td>5500</td>
<td>Kg</td>
</tr>
<tr>
<td>Couple</td>
<td>C giunto</td>
<td>4450</td>
<td>Kg*m</td>
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<th>Characteristic</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
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<td>Nominal Couple</td>
<td>C nom</td>
<td>3363,53</td>
<td>Kg*m</td>
</tr>
<tr>
<td>Radial load</td>
<td>F rad</td>
<td>4418,11</td>
<td>Kg</td>
</tr>
<tr>
<td>Couple Verification</td>
<td></td>
<td>0,75585</td>
<td>76%</td>
</tr>
<tr>
<td>Load Verification</td>
<td></td>
<td>0,80329</td>
<td>80%</td>
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Calculation procedure

Two main parameters are calculated in the Couplings Design procedure:

1- Radial force exerted on joint:

\[ F_{radiale} = (F_0 \times \frac{nd}{9.81} + m_{drum})/2 \]

2- Nominal torque of joint:

\[ C_{nom} = \frac{SF \times Dp \times P \times nt}{2000 \times ng \times T} \]

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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