POLITECNICO DI TORINO

DIMEAS - Department of Mechanical and Aerospace Engineering

Master of Science Degree in Automotive Engineering

Master's degree thesis

Multi-positioning Test Bench Design for Manual Gearshift Command with Implementation of Active Reaction Force

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Acknowledgements

I would like firstly to thank my family, for the support they have given me over this years; my parents, who taught me to face life and allowed me to perform university studies, believing in me. I would like also to thank my girlfriend, always next to me, even in most difficult moments of this experience, with her smile.
Thanks to my colleagues, available at all times to help and advise me at the beginning of my career path.

Summary

Starting from XIX century, the automotive sector, got a huge development with a deep influence on people's lifestyle. The improvements reached nowadays, in terms of performances, require a strict control of each component and a testing activity to guarantee their reliability. This project of thesis has been developed at Sila Holding Industriale SpA, one of the most important global companies in the production and distribution of gearshift system and flexible remote control cables. The customization of the product, according to customer's specifications, requires a Testing department, to obtain a robust prototyping and a reliable final product. The aim of the Thesis is the development of a new test bench, to overcome the limits existing on actual solutions. The work followed five steps:

- Study of the gearshift system.
- Study of existing test benches: analysis of their working principles, advantages and drawbacks.
- Development of a new solution: design and evaluation of multiple alternatives through a CAD software.
- Realization of designed solution.
- Validation of designed solution.

The design in particular was focused on a new Actuation system and a new Active Reaction Force system.

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Chapter 1

Introduction

To supply a robust product, every OEM, needs a Testing department. The objective is to guarantee the conformity of components respect to customer's requests. To meet the increasingly stringent needs, of the automotive sector, the testing has become a core activity in the product development, and it is involved all along the production process:

- Prototyping: During the first steps of the product development, a prototype with provisional solution is produced. The prototype is used to display to the client the solution designed by the company. Often the additive manufacturing is exploited in the prototyping, due to its multiple advantages: reduced cost, no mould required, time saving, flexibility.
  On the other side, to get a preliminary validation of chosen design, is required to produce a prototype through a manufacturing process akin to the final one.

- Design Validation: After the design of the gearshift system, the customer requires a Design validation, which has the task to confirm the target established. The test engineer carries out internal test, to confirm that the products comply with manufacturer requirements (see section 2.3.3).
  The necessary modification, on the product design, are carried out, until it respects completely customer's targets. At this point the design validation can be considered completed; actually minor modification can be implemented, in following steps.

- ADVP&R: Once completed the Design Validation, is required a Validation of the Process (Analysis Development Validation Plan & Report). Even at this point, the testing department checks that the industrial process, at the End of Line, is able to obtain a reliable product, which undergoes targets achieved during the previous step. Modifications on the production of single components, or on the assembly methodology, are put into effect, requiring a validation for each variation executed. The ADVP&R is a process of planning, testing and reporting, performed to verify that components meet a set of requirements established by engineers, during design phase. Usually an outlining of the test required is reported on a spreadsheet, the components have to fulfil the tests targets to be approved for the production process of the OEM. This process is useful to detect failures mode and get reliable products.

- Re-qualification: In agreement with the customer, is required a re-qualification, usually after 2 years. The reason is to confirm that production's standards are respected, and a dispersion in the known limits has not occurred. To complete this point, the testing department has to repeat a list of tests, like in Design Validation phase.

Chapter 2

State of art

In this chapter, is presented an overview of the existing test benches. The principles of functioning, the advantages and drawbacks of the systems, currently used in the company, represents the starting point to design a new solution, which overcome the existing limits. Additionally is provided an explanation of the components produced and tested in the company, to make clear the objectives of the tests.

2.1 Test Bench

Depending on the request and the objectives, a different test bench is designed specifically, to test the gearshift system. This thesis, is focused on the design of a test bench, with the purpose of Characterize the system and perform Durability test (see section 2.3.3). The three elements that made up a Test Bench are: Fixture, Actuation system, Reaction system. The fixture is designed to reproduce vehicle routing position, on every bench, to obtain results closer as much as possible to a real condition. The customer provides the layout, the reference points and mandatory parameters of the system. These data are useful to designers, to produce a component that fully respects the requirements.

2.1.1 Actuation System

The actuation systems currently exploited by the company are two:

- Rotary actuation
  This equipment, shown in Figure 2.1, exploits a double rack pinion system. Rack-pinion 1 (Fig. 2.2) actuates shift motion, rack-pinion 2 (Fig. 2.3) actuates select motion. The motion, and the control, of the rack pass through a cable actuation. The cable-ends of the control cables, engage on one side the rack

on a sphere (indicated in figure with a red circle), on the other side a sphere onto an electric motor (Fig. 2.4).

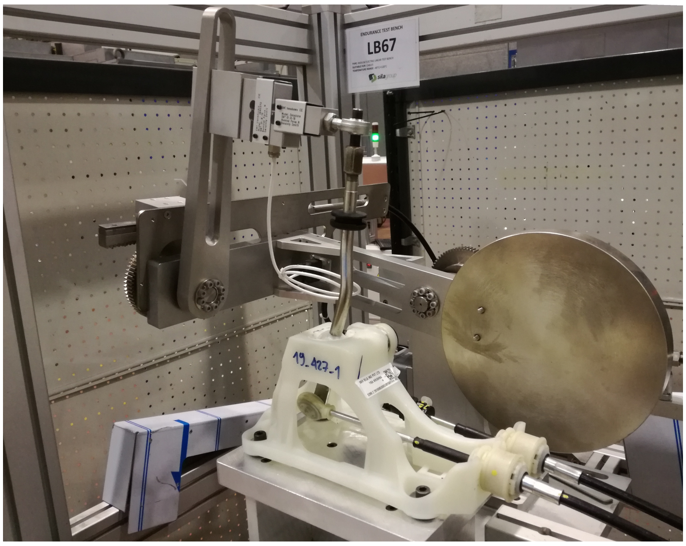


Figure 2.1: Rotative Actuation

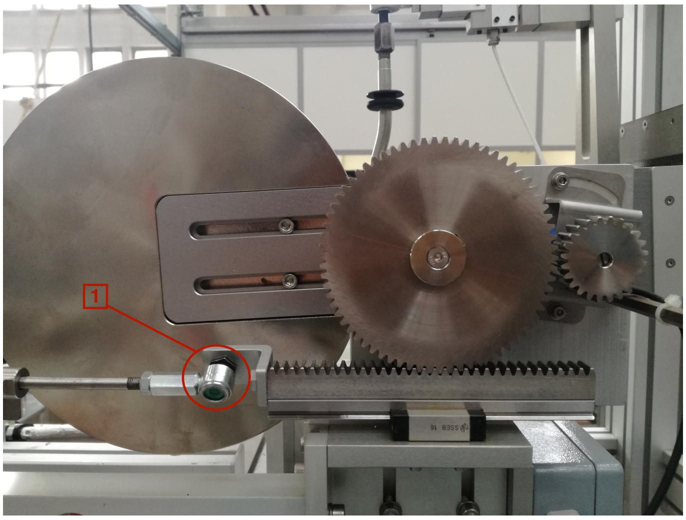


Figure 2.2: Rotative Actuation: Rack-pinion 1

The brushless electric actuator, Figure 2.4, has the properties reported in table 2.1.

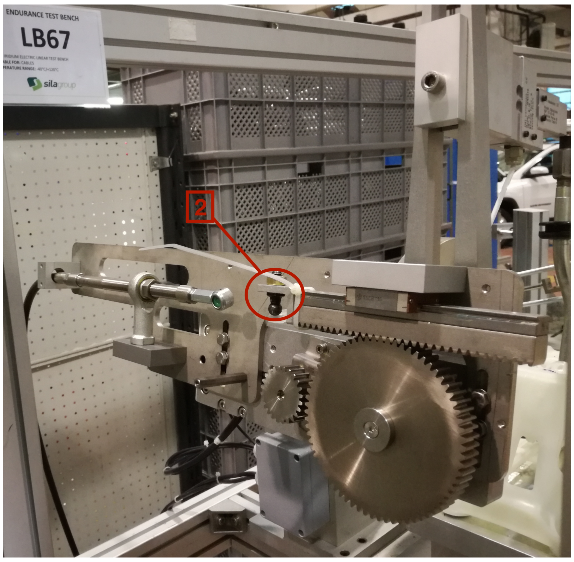


Figure 2.3: Rotative Actuation: Rack-pinion 2

Table 2.1: Cables actuation: Actuators characteristics

Working voltages	48 V
Maximum Load	600 N
Maximum Stroke	$\pm 75\ mm$
Maximum Speed	10 mm/s
Working Temperature	$-5°C \sim +155°C$

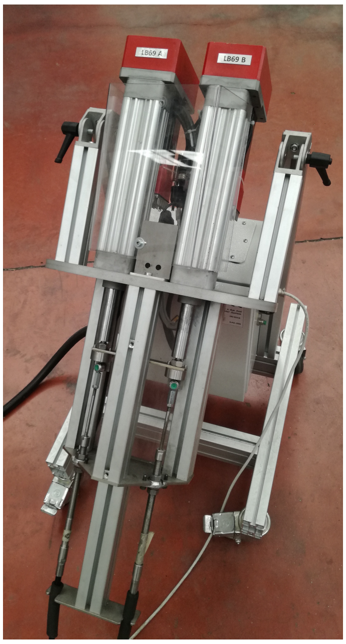


Figure 2.4: Brushless Electric Actuator

The advantages of this system are: good packaging, wide temperature variation supported.
The drawbacks of this system are: high cost, high inertia.

- Cartesian actuation
  This equipment, shown in Figure 2.5, exploit two brushless motors, positioned in a perpendicular layout on the same plane. Both actuators are protected by a insulation case, which allows the test bench to work in a climatic cell (the actuators can be cooled by air). E.m. motors characteristics are reported in Table 2.2.

Table 2.2: Cartesian Test Bench: Actuators characteristics

Working voltages	48 V
Maximum Load	300 N
Maximum Speed	$\sim 1\ m/s$
Working Temperature	$-40°C \sim +130°C$
Cost of Test Bench	50000 €
Dimensions of Test Bench	1800x1400 mm

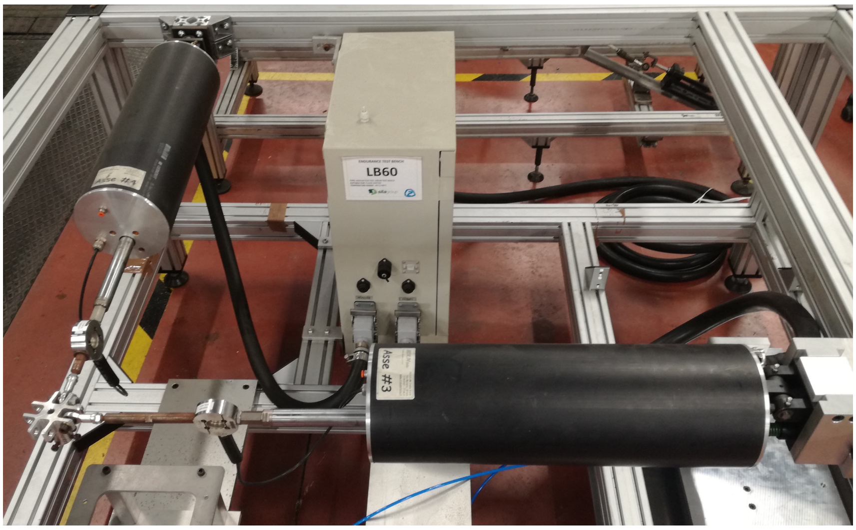


Figure 2.5: Cartesian Actuation

The advantages of this system are: high speed achievable, high force achievable, wide temperature variation supported.
The drawbacks of this system are: high cost, very low packaging, force application on single plane, air cooling at high temperature.

2.1.2 Reaction System

Currently only one system is adopted as reaction, shown in Figure 2.6. The system exploits a coil, usually characterized in accordance with customer's request. This kind of reaction allows a regulation of the strokes; the coil can be pre-charged, through a regulation with two metal plates. The cables act on a rod, which is able to slide through two linear bearings inside metal blocks 1 and 2. Metal plates 3 and 4 are screwed on threaded rods, locking and regulating the coil. A load cell positioned between cable-end and main rod allows the valuation of the force applied in compression and extension. Advantages of the system: low cost, low maintenance required, wide range of working temperature, adaptability to different targets (switching the coil).
Disadvantages of the system: fixed relation force-stroke, difficult correct setting (the alignment of the system depends on 8 points, indicated with blue points on Fig. 2.6, which deeply influence the correct functioning).

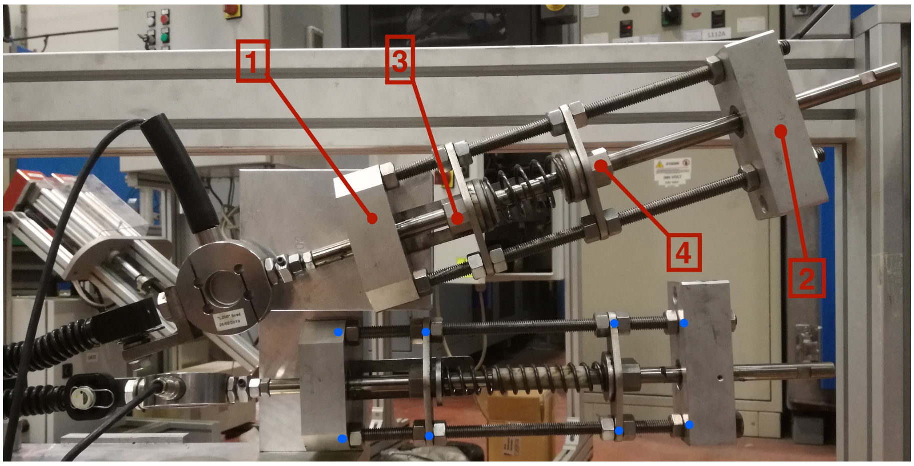


Figure 2.6: Reaction System

2.2 Software

The software used to control the test bench, and to perform the tests, is WinData (released by IRIDIUM Italia S.a.s.). The software bases its control on a feedback scheme, which exploits data from load cells and potentiometers. To control the test bench, WinData, has the interface showed in Figure 2.7 and in Figure 2.9, it allows two kind of setting:

- Characterization: to characterize a component (see Subsection 2.3.3), the extension of the software used is WinData SUITE!. The program allows to set the desired number of steps, and in each step a control target, based on force or stroke. Other parameters: the direction of actuation, the speed of actuation and a safety threshold which interrupts the test. The software can rework the sample collected and modify the graphs, in accordance with the setting chosen by the user.

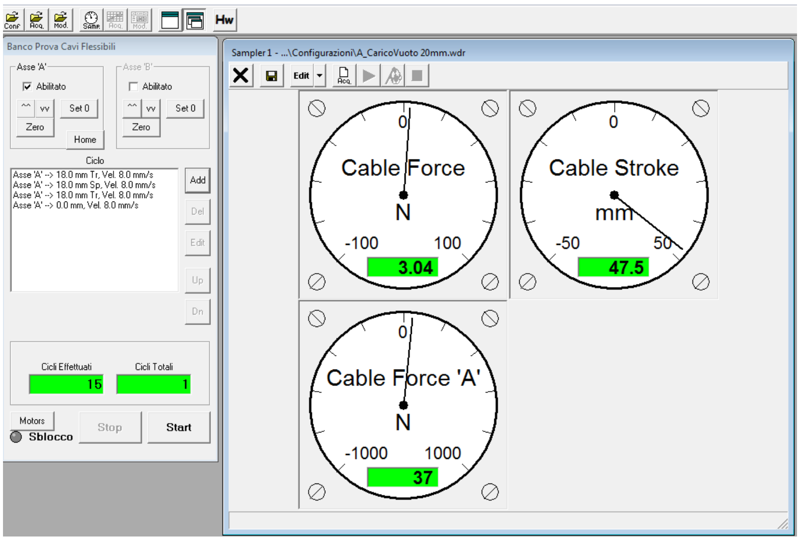


Figure 2.7: WinData SUITE!: Characterization Setting

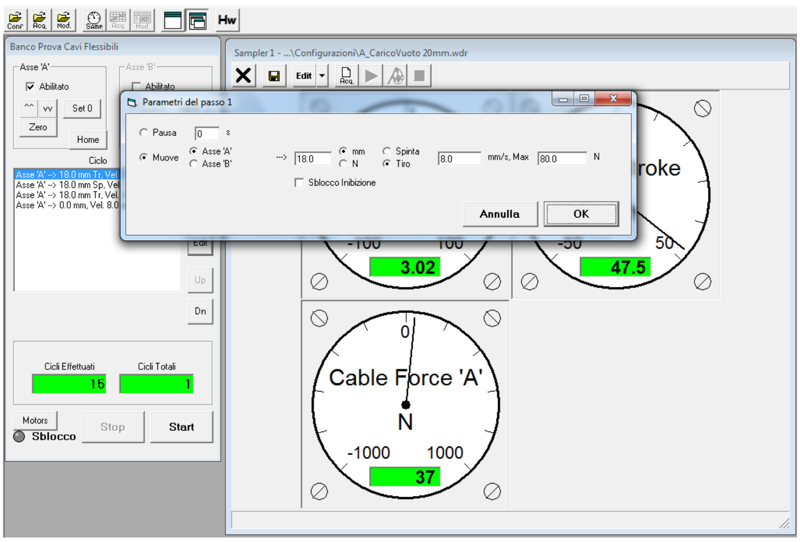


Figure 2.8: WinData SUITE!: Characterization Edit parameters

- Durability: to perform a Durability test, the extension WinData LongRun is exploited. The program distinguishes the component tested (Complete Gearshif system, Only Cables, Only Shifter) and the kind of test bench used. The setting comprehends: the number of cycles (which can be different on the two axis), the target of force or stroke in each cycle, the frequency of the cycle, safety threshold based on the force reached (recognizes failures of the system), climatic cell control.

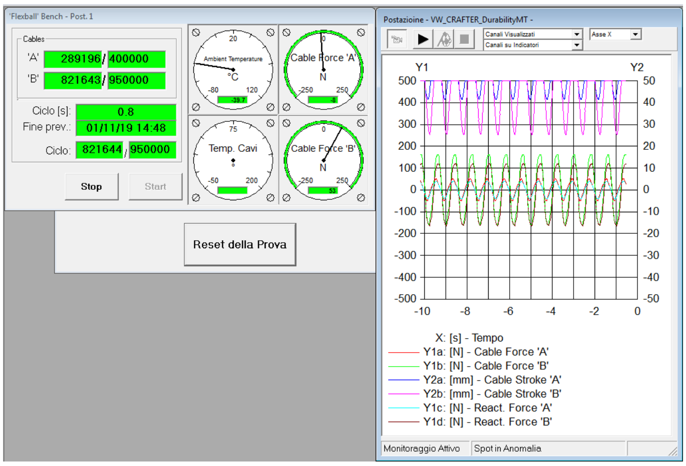


Figure 2.9: WinData LongRun: Durability

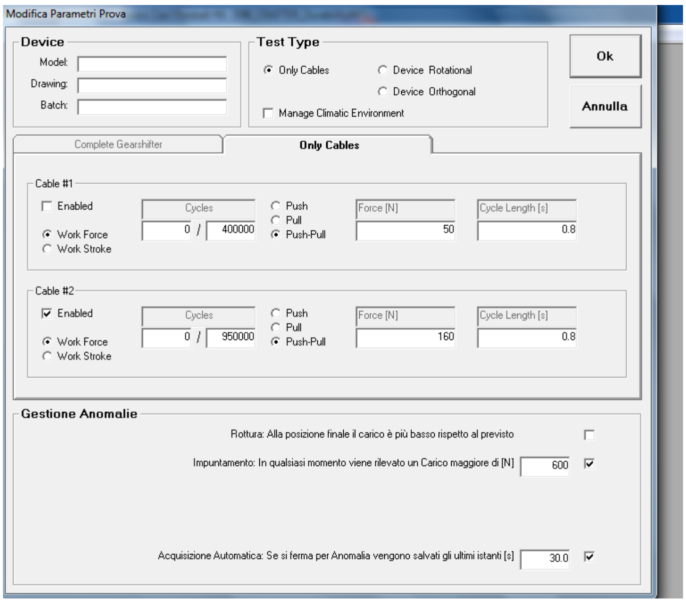


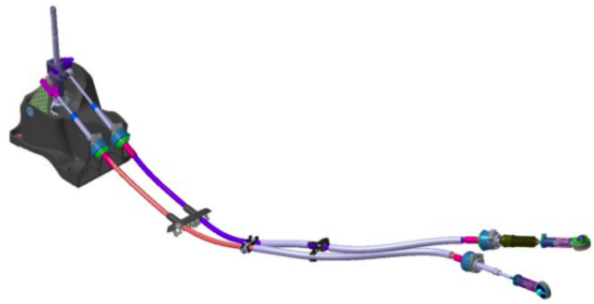
Figure 2.10: WinData LongRun: Durability Setting

2.3 Gearshift System

The gearshift system produced by Sila can be divided in 3 main categories:

- Manual gearshift system
  This kind of system is identified in three different mounting in Fig 2.11: Manual underfloor mounted gearshift system, Manual overfloor mounted gearshift system, Manual dashboard mounted gearshift system.

(a) Underfloor mounted



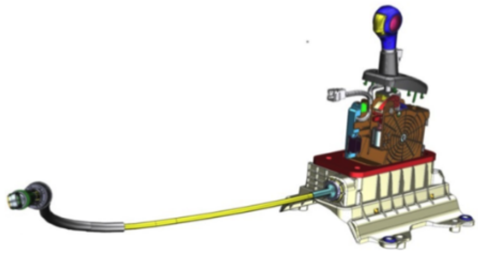
(b) Overfloor mounted

(c) Dashboard mounted

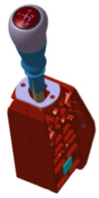
Figure 2.11: Manual Gearshift System

- Automatic gearshift system The Automatic gearshift system can be classified depending on functioning principle (Fig 2.12): Shift and Park by Cable, Shift by Wire - Park by Cable, Shift and Park by Wire.

(a) Shift and Park by Cable



(b) Shift by Wire - Park by Cable



(c) Shift and Park by Wire

Figure 2.12: Automatic Gearshift System

- Electronic gearshift system

The system, used as reference, in the development of this project, is a complete Manual Overfloor Mounted Gearshift System for SUZUKI project YBA. The two main component of the system are the Shifter and the Cables.

2.3.1 Shifter

The Shifter is composed by multiple parts, each fundamental for the correct functioning of the system. An overview on three main components is here presented.

- Main Housing
  This component represents the structural part of the system. The materials exploited typically are: PA 6.6 GF35, PA 6.6 GF50, PP GF50. The design of the main housing must comply with specific customer's requirements, the main parameters, which affect the part are: Packaging, Overfloor/Underfloor/Dashboard mounting, Ergonomic targets (from customer), Material, Cost, Structural strength, Integrated components. The last point mentioned, refers to the possible integration of components, usually co-moulded, which allows to reduce the final cost of the system. When allowed, the customer, approves this kind of integration, in the main housing mould, to reduce number of parts, reduce the cost and improve the structural strength of the overall system. In

figure 2.13 are reported some example of main housing, and the indication of possible integration in it.

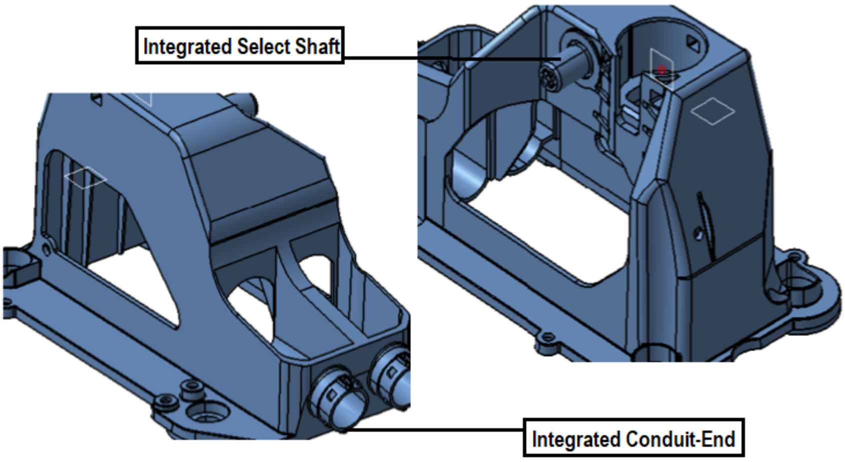


Figure 2.13: Main Housing - Integration example

Other elements visible in Fig 2.14 are the Inhibition Barrier (solution adopted to prevent accidental reverse gear engagement), the Seats for rubber bushings (solution adopted to damp down main housing motion, positioned in the coupling point with the vehicle), the Conduit-End interface (solution adopted to fix, or remove, rapidly the conduit-end on the main housing), the Seat for Main Lever Sphere.

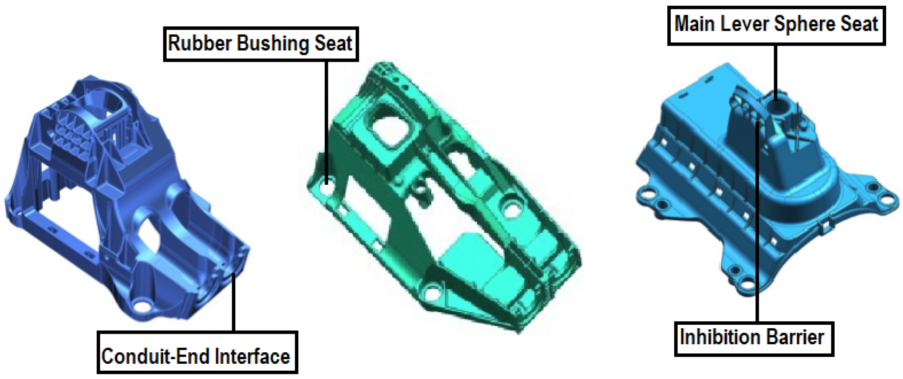


Figure 2.14: Main Housing - Solution example

- Main Lever
  The main lever may appear as a very simple component, nevertheless it recalls multiple function:

- Transfer driver's force to the cable
- Engage gears
- Engage reverse gear
- Limit cables stroke

The practical solution, adopted in the design of this component, are almost unlimited, customized for each client, and affected by a trade off between:

- Materials
- Cost
- Strength
- Integrated parts
- Knob type
- Reverse gear inhibition system
- Fulcrum concept
- Locking device
- Start & Stop System (Neutral position detection)

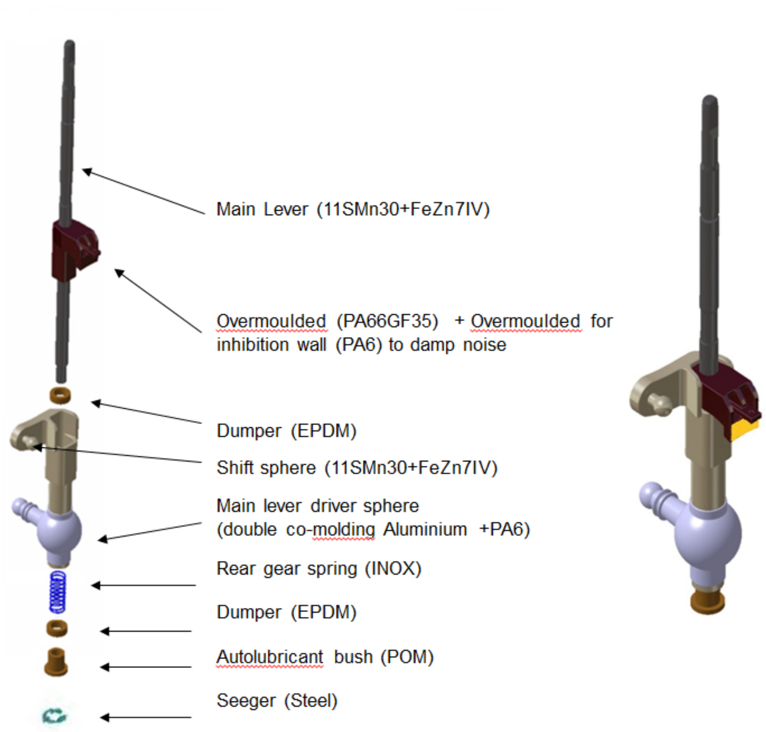


Figure 2.15: Main Lever - Solution example

The co-moulding, is always requested, in the production of the main lever, so the technology available to the OEM, and the cost accepted by the customer (depending on the material), deeply influence the final result.
Some solution are here displayed, to get a plainer idea of the entire system.

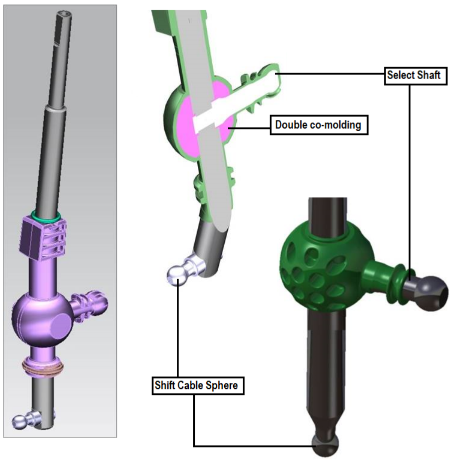


Figure 2.16: Main Lever - Solution example

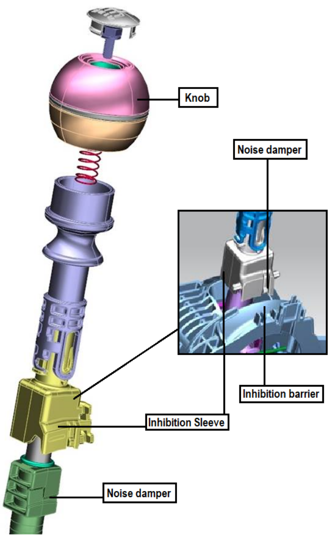


Figure 2.17: Main Lever - Solution example

- Select Lever & Select Spring
  Select spring and lever together allow to transfer, the force applied by the driver in y direction, along x direction. The change in force direction application,

passes through a torsion spring (Fig 2.18), which has its reaction point on select shaft or on a specific device. The design of the assembly, defines a transmission ratio, from the driver to the cables, which multiplies the force at the input, and allows the gearshift actuation.

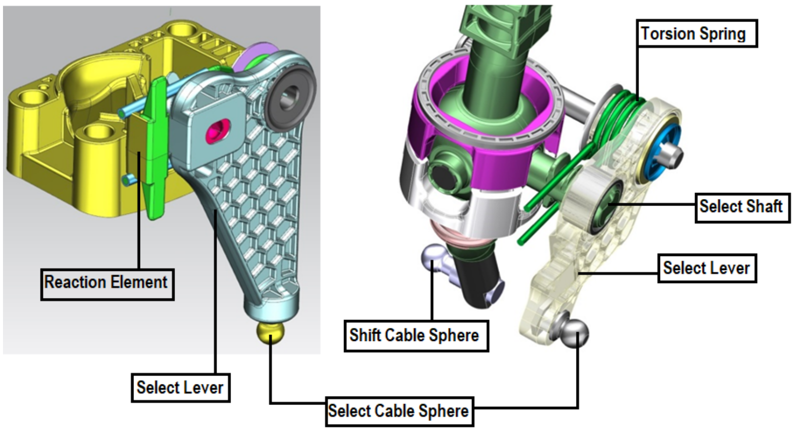


Figure 2.18: Select Lever - Solution example

Typically the material exploited is PA6 GF35, with co-moulded metal sphere (to move the select cable) and metal or plastic bush (to host the select shaft). Despite the final shape of the lever, the concept behind the force transmission doesn't change, from one solution to another. Even for this component, depending on force involved in the system, and the cost accepted by the customer, a trade off defines the technical solution adopted.

The technical solutions adopted on Suzuki YBA Shifter, are shown in fig. 2.19. All the main components previously described, are visible. The shifter is a Manual Overfloor mounted kind. Are visible the sphere to engage the shift cable-end, on the main lever, and a cylindrical pin to engage the select cable-end on the select Lever. In figure 2.20 are visible the 4 seats for rubber bushings and the frontal seats for quick-fit of the cables.

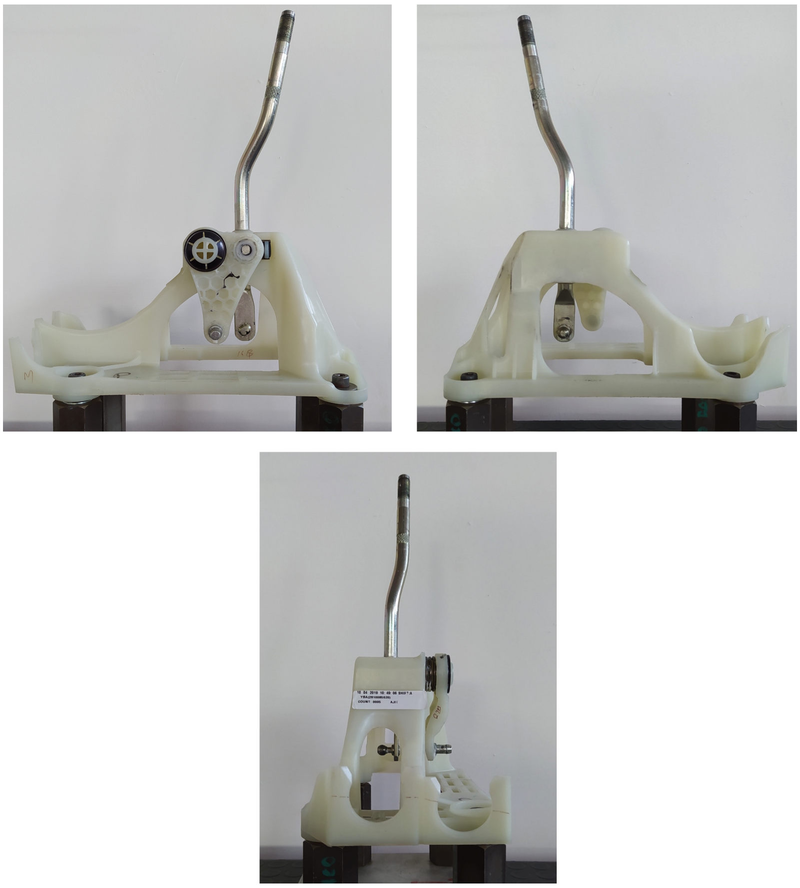
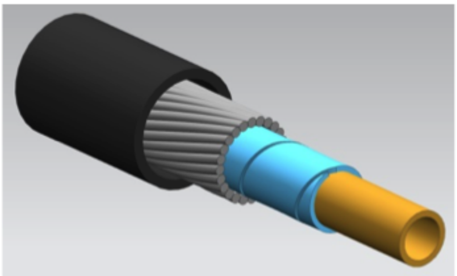


Figure 2.19: Suzuki YBA Shifter

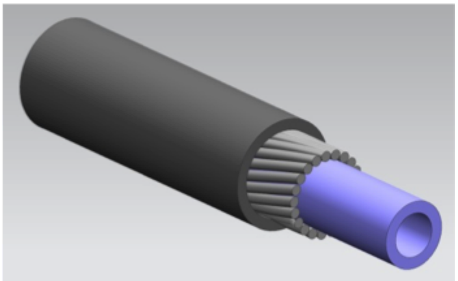
Figure 2.20: Suzuki YBA Shifter

2.3.2 Cables

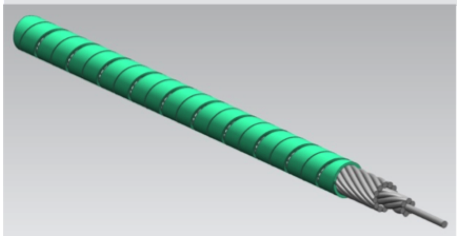
The cables adopted in a gearshift system, are of push-pull kind. The technology behind push-pull cables, exploits steel wires (and plates), which combined in a sufficient number, are able to transmit the force necessary to engage gears and to support huge loads.
In this section, is presented an overview of the main typology of cables, produced nowadays by Sila.



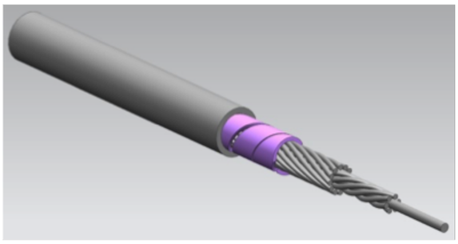
- External diameter 9.5 mm or 10.1 mm
- Internal diameter 3.4 mm or 4.25 mm
- Liner in PTFE or PBT
- Flat wire Zinc Plated Steel
- 24 wires ZINC PLATED STEEL
- Plastic coating PA6 or HYTREL



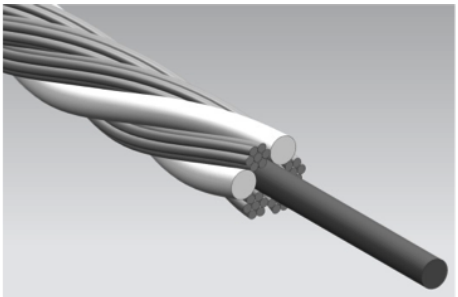
- External diameter 8.5 mm
- Internal diameter 3.45 mm
- Liner in PBT
- 21 wires ZINC PLATED STEEL
- Plastic coating PA6 or PP



- External diameter 3,2 mm or 4 mm
- 1 inner cable ZINC PLATED STEEL
- 18 or 27 Wires ZINC PLATED STEEL
- Double flat wire STAINLESS STEEL or Plastic coating PA12



- External diameter 2.44 mm or 3,2 mm
- 1 inner cable ZINC PLATED STEEL
- 18 Wires ZINC PLATED STEEL
- Double flat wire STAINLESS STEEL
- Plastic coating PA12 (or w/o coating)

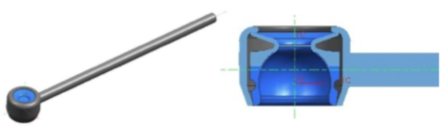


- External diameter 3,2 *mm*
- 1 inner cable ZINC PLATED STEEL
- 2 Wires ZINC PLATED STEEL
- 4 Strands (each strand with 7 wires)

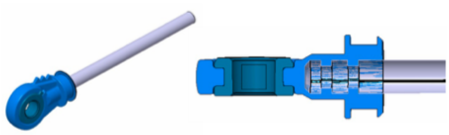
The introduction of the push-pull cables, brought huge advantages in the gearshift mechanism exploited by vehicles. The main reasons, which define the success of this technical solution, respect to the previous one, are:

- Weight
- Flexibility
- Packaging
- Resistance to oxidation
- Vibration filtering

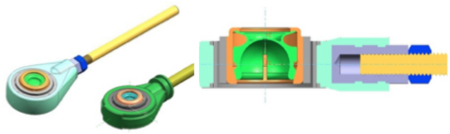
The second important element of the cables, is the Cable-end. This component is exploited to fix the cables on one side on the gearbox's leverage, on the other side on the shifter (on main lever or select lever, see section 2.3.1). Along the years, Sila developed a wide variant of cable-end, here are presented the main solutions adopted, to have a complete overview of the system described.



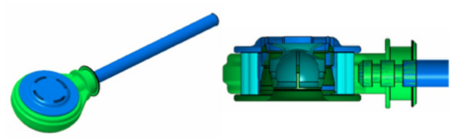
- Spherical rigid cable end (4 versions)
- Metallic case
- Sphere diameter 10 *mm* or 13 *mm*
- Pull out load > 3000 N



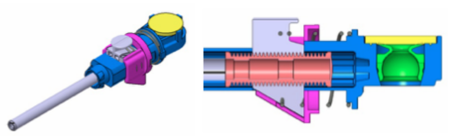
- Semi-rigid cylindrical cable end (2 versions)
- Sphere diameter 8 *mm*
- Pull out load > 2000 N



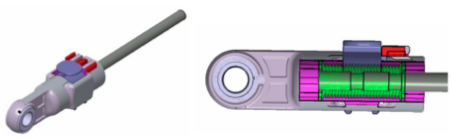
- Spherical damped cable end (5 versions)
- Available in adjustable version
- Over moulded plastic case
- Sphere diameter 10 *mm* or 13 *mm*
- Pull out load > 3000 *N*
- Stiffness according to customer request



- Spherical damped cable end (5 versions)
- Over moulded plastic case
- Sphere diameter 10 *mm*
- Pull out load > 2000 *N*
- Stiffness according to customer request



- Compact spherical rigid quick adjuster
- Sphere diameter 10 *mm*
- Adjustment range ±6 *mm*
- Pull out load > 2500 *N*
- Clearance recovery based on spring position



- Semi-rigid cylindrical quick adjuster
- Sphere diameter 10 *mm*
- Adjustment range ±6 *mm*
- Pull out load > 2000 *N*

An additional component, exploited to fix the cable in vehicle routing position, is the Conduit End, differentiated between rigid or damped. Two example are shown in fig. 2.21 , the right one is mounted on Suzuki YBA cables.

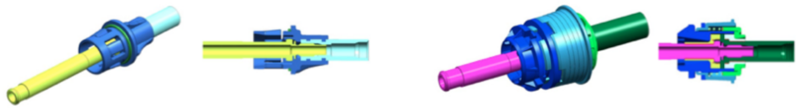


Figure 2.21: Conduit End - Example

The cables adopted in YBA project are shown in fig. 2.22 and 2.23. The shift cable exploits on both sides a spherical damped cable end, often a damped solution is required on shift side. The select cable instead adopts on main lever side a semi-rigid cylindrical quick adjuster, while on gearbox side a semi-rigid cylindrical cable end. The quick adjuster solution, offers the possibility to adapt the length of the cable, and as consequence the position of the main lever, to the gearbox condition.

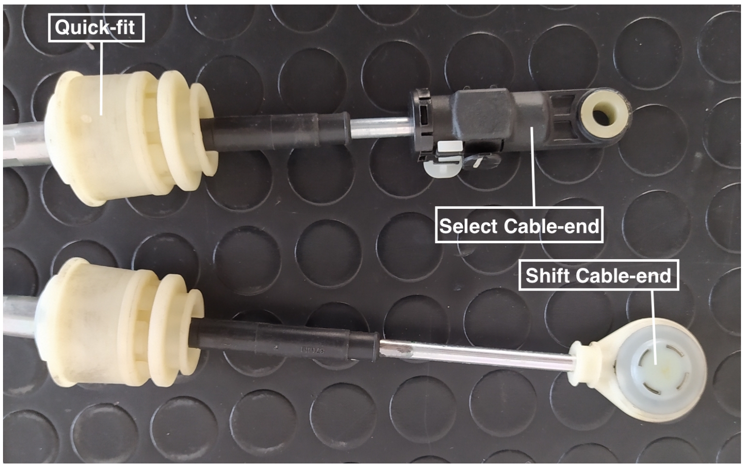


Figure 2.22: Cable-end Shifter side

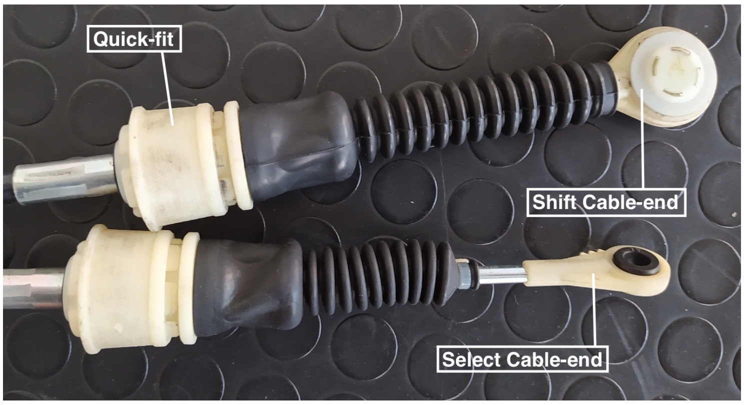


Figure 2.23: Cable-end Gearbox side

2.3.3 Tests

A complete description of all the tests performed, will require a too extensive analysis. For the aim of this thesis, is exposed an overview of the tests of interest, which are also the one performed, always, for every manufacturer. In addition to the test explained in the following, every customer, requires a certain number of test, depending on the characteristics of the system, and the features considered weaknesses; often those additional tests need a specific test bench.

- Freeplay: the measurement can be performed on the complete system, or on single component (Shifter, Cables). The test checks the freeplay of the affected parts, with the output locked, to measure a value referred only to the gearshift system. Focusing on YBA project, the freeplay is requested only on the shifter, the target value defined by Suzuki are reported in table 2.3.

Table 2.3: Freeplay target specification - Suzuki

SHIFTER	TARGET
Shift direction $\pm 3\ N$	$\leq 0.5\ mm$
Select direction (w/o Select spring) $\pm 3\ N$	$\leq 1\ mm$
Select direction (with Select spring)$\pm 3\ N$	$\leq 2\ mm$

- Internal Friction: the measurement can be performed on the complete system, or on single component (Shifter, Cables). The test checks the force required to actuate the affected component. The cable end on gearbox side are free, the force contribution measured depends only on internal friction of the component. Focusing on YBA project, the target are reported in table 2.4.

Table 2.4: Internal Friction target specification - Suzuki

SHIFTER	TARGET
Shift direction $\pm 30\ mm$ cables stroke	$\leq 2\ N$
Select direction $\pm 13\ mm$ cables stroke	$\leq 7\ N$
CABLES	
Shift direction $\pm 30\ mm$ cables stroke	$\leq 5\ N$
Select direction $\pm 13\ mm$ cables stroke	$\leq 5\ N$
SHIFTER + CABLES	
Shift direction $\pm 30\ mm$ cables stroke	$\leq 8\ N$
Select direction $\pm 13\ mm$ cables stroke	$\leq 7\ N$

- Efficiency: the measurement can be performed on the complete system, or on single component (Shifter, Cables). The test checks the difference of load from input to output, measuring the losses along the system, and defining the efficiency. To obtain measurement, which reflects a real condition, the vehicle layout is reproduced. Target value required by Suzuki on YBA project are reported in table 2.5.

Table 2.5: Efficiency target specification - Suzuki

SHIFTER	TARGET
Shift direction ± 100 N	≥ 90 %
Select direction ± 100 N	≥ 80 %
SHIFTER + CABLES	
Shift direction ± 100 N	≥ 80 %
Select direction ± 100 N	≥ 75 %

- Limit Tensile Pressure: the measurement can be performed on the complete system, or on single component (Shifter, Cables). The objective, is to test the structural strength of the system, applying an high load, in a specified direction, with the output locked. On YBA project the target value are the following.

Table 2.6: Limit Tensile Pressure target specification - Suzuki

SHIFTER	TARGET
Each direction at RT $20°C$	600 N
CABLES	
Shift direction (each gear) at RT $20°C$	600 N
Select direction at RT $20°C$	600 N
SHIFTER + CABLES	
Shift direction (each gear) at $-30°C; 20°C; 80°C$	600 N
Select direction at $-30°C; 20°C; 80°C$	600 N

- Durability: the test can be performed on the complete system, or on single component (Shifter, Cables). To obtain a result, which reflects a real condition, the vehicle layout is reproduced. SUZUKI specifications for this test are reported in table 2.7, and the thermal cycle in figure 2.24. The requirements are listed as follow:
  - Shift direction:
    - Must not be found crack and destruction
    - No noise during test

    * Efficiency after test > 90%
    * Clearance increase after the test < 0.5 *mm*
    * Freeplay increase after the test < 2 *N*
- Select direction:
    * Must not be found crack and destruction
    * No noise during test
    * Efficiency after test > 80%
    * Clearance increase after test < 2 *mm*
    * Freeplay increase after the test < 7 *N*

Table 2.7: Durability target specification - Suzuki

SHIFTER + CABLES	
Shift direction	$\pm 100\ N\ at\ \pm 30\ mm$ (cable stroke)
$1st \leftrightarrows 2nd$	400000 cycles
$3rd \leftrightarrows 4th$	400000 cycles
$5th \leftrightarrows Rev$	200000 cycles
Select direction	$\pm 100\ N\ at\ \pm 20\ mm$ (cable stroke)
$Low\ side \leftrightarrows Neutral \leftrightarrows Rev\ side$	500000 cycles

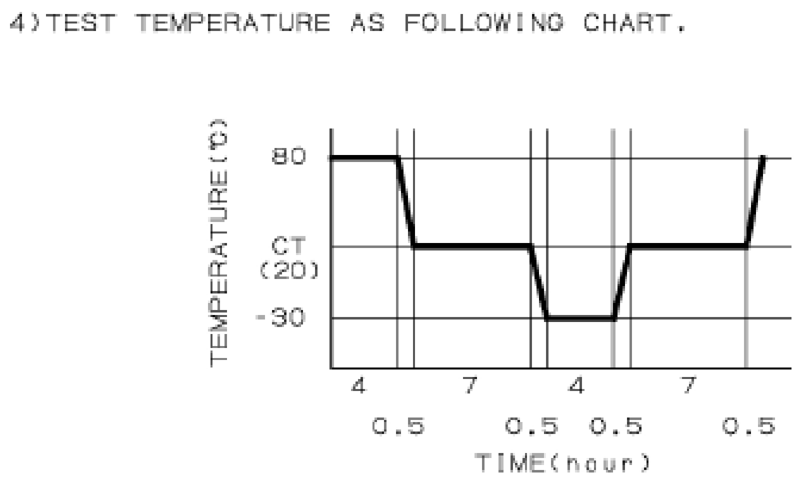


Figure 2.24: Durability Thermal Cycle - Suzuki

Chapter 3

Objective of the Thesis

To go over the existing limits of test benches, presented in section 2.1, this work of thesis focus on the design of a new system. The aim, is the actuation of the gearshift system, which works together with a reaction system, to perform the testing activity, described in section 2.3.3. The main limits of actual systems are: cost, size, inertia, reliability, speed. The specific requirements of the new actuation system are:

- Low inertia of the system
- Low cost
- Robustness
- Flexibility
- Reliability
- Compact
- High speed (up to 12 mm/s)
- High frequency (up to 1.25 Hz)
- Resistance to wide range of Temperature $[-40 \div 100]$ $(^\circ\ C)$
- High efficiency
- Actuation Forces up to 200 N

For what concern the reaction system, the main problem is related to a fix reaction force. The requirements, in durability test (see 2.7), specify a relation force-stroke. The limit of the actual system, consist in a regulation force-stroke, which satisfies the requirements when the test start. However during the test, the following changes occur:

- High temperature: the system is more compliant, a constant force produces longer stroke.
- Low temperature: the system is more stiff, a constant force produces shorter stroke.
- High number of cycles: clearances in the system increases, efficiency of the system decreases. Relation force-stroke changes along the test.

The points just mentioned, are not in accordance with a reaction system that works on a fix regulation force-stroke. A second drawback, of the actual solution, is its regulation method. The position, and pre-load of the spring, is affected by the alignment of 8 points (fig. 2.6). Without the correct setting of the test bench, the reaction system works incorrectly, requiring the application of higher force, to produce the desired stroke. In this work of thesis, is presented an alternative solution, to be implemented on test bench. The new system must have:

- Resistance to wide range of Temperature $[-40 \div 100]$ $(^\circ\ C)$
- Active reaction
- Low cost
- Compact size
- Reliability
- Reaction Force up to 600 N

Chapter 4

Test Bench Actuation Design

4.1 Actuation System - Solution 1

Figure 4.1: Actuation System - Solution 1

The first solution developed, is presented in fig. 4.1. The idea, is to exploit the the electric motor, in fig. 4.2, with the push-pull cables. The aim of this choice, is to decrease the cost of the new test bench:

- 12 Electric Motor (couples) available
- 12 Control Workstation available
- Control Software available

- Push-Pull cables produced internally

The electric motors, already available in the company, result to be an effective and reliable energy source, for the actuation of the Test Benches. Moreover the software for their control, WinData, is already implemented on multiple workstations, and is not required the training on a new one.

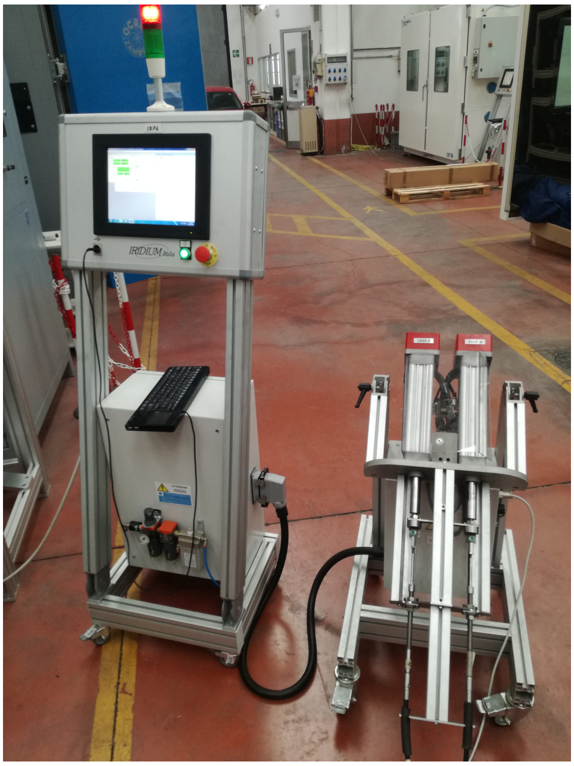


Figure 4.2: Workstation - Electric Motors

The two motors are used to control independently and simultaneously, the shift and select motion. To perform this parallel control, the idea is to exploit two rack-pinion systems. Rack-pinion 1, actuates a piston rod system, which acts on Shift direction. The application of piston-rod in shift direction, has been chosen, as the force's requirements in this direction are higher. The rod acts on select system, fig. 4.3, which constrains the translations, leaving only x translation as DOF.

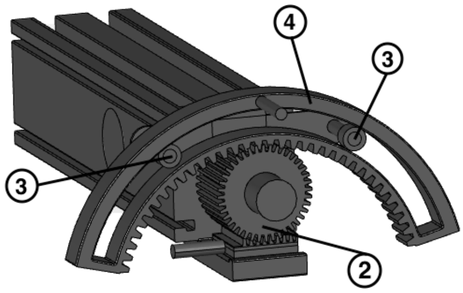


Figure 4.3: Solution 1 - Select System

In addition to the requirements, exposed in chapter 3, an optional features is the rotative motion. Often customers require, in principle, an actuation which performs a rotative motion, to reproduce the arch of the knob. On actual Cartesian test bench, fig. 2.5, the actuation motion is linear. To promote the height change, in the different points of the arch, springs are exploited, so the actuator is able to rotate around its hinges.
In the first solution, the rack-pinion 2, actuated by electric motor, transfers the linear motion of the rack, to an arch 4. The arch is an internal gear, which rotating on bearings 3 acts directly on the main lever. With a correct dimensioning of the arch, is possible satisfy the rotative motion's request on zy plane.
The advantages of this solution are mainly in shift direction: Robust, High force transmission, High efficiency, Resistance to wide range of temperature. Nevertheless the drawbacks are multiple:

- Inertia of Select system translation
- Cost of the Internal Gear Arch
- Size of the complete system
- Structural strength of the Select system
- Practical - Technical solution in Select system constrain to translation

The listed points, are the reasons to reject the Solution 1 as new actuation system.

4.2 Actuation System - Solution 2

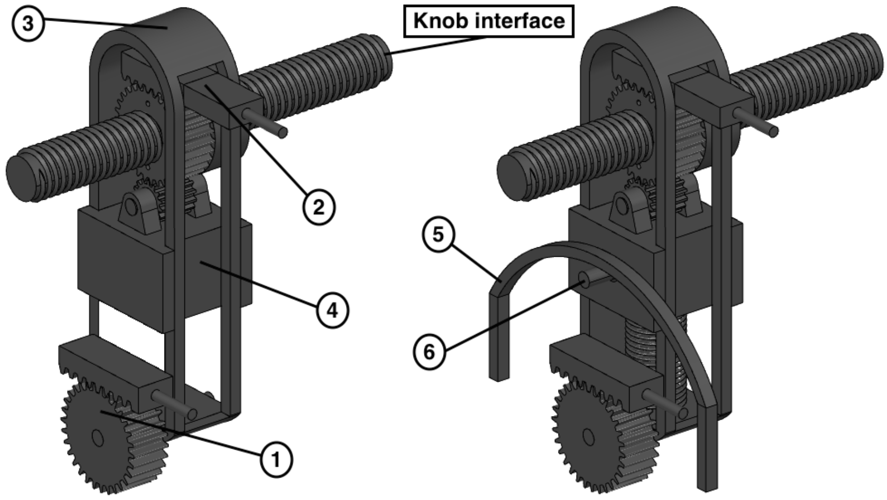


Figure 4.4: Actuation System - Solution 2

The second solution developed, is shown in fig. 4.4. To maintain low cost, as in the first solution, the electric motors already in use are exploited, and so a double rack-pinion system. Rack-pinion 1, engaged with oscillating body 3, when actuated brings all the system to swing, with the motion of the knob in select direction.
The shift system, fig. 4.5, is mounted on body 4, here two alternatives are proposed in fig. 4.4, to regulate the height respect to main lever length. On the left, alternative 1, body 4 can be regulated, screwing it on oscillating body 3. On the right, alternative 2, a spring pushes body 4, which through the pin 6, rotates at the distance imposed by arch 5 diameter. This alternative allows the shift body to maintain always the correct height, performing the same arch of the knob. The substitution of the arch, allows to adapt the system to any gearshift system. The advantage of the alternative 1, is to avoid the manufacturing of the arch, with a fixed regulation.
In figure 4.5, can be seen a detail of the shift system. The rack-pinion system 2, works together with a ball-screw system. Gear 7 receives tangential force from rack, and transfer the moment to component 8. The moment is converted in a translation along x of component 9, and so of the knob.

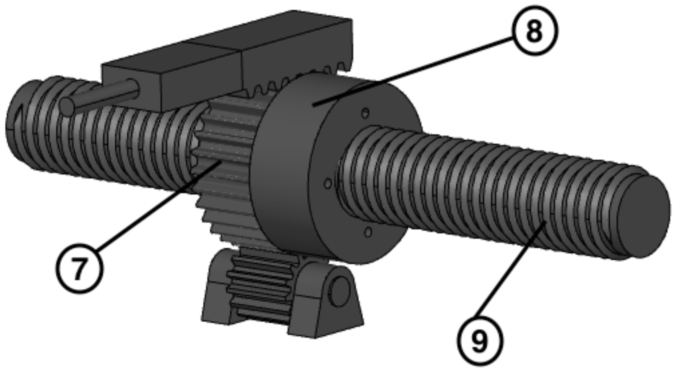


Figure 4.5: Solution 1 - Shift System

The advantages of the system are:

- Low inertia.
- High Flexibility.
- Low cost.
- Low number of components.
- Reliability.
- Resistance to wide range of Temperature.
- Compact size.

The disadvantages: Efficiency and Speed achievable.
The drawbacks stand mainly is the ball-screw system. This technical solution has low efficiency, and would require a too high linear translation of the rack, to guarantee the sufficient translation of the component 9. An additional problem is the speed achievable in shift direction, respect to the requirements, not sufficient if ball-screw is actuated by a rack-pinion.

4.3 Actuation System - Solution 3

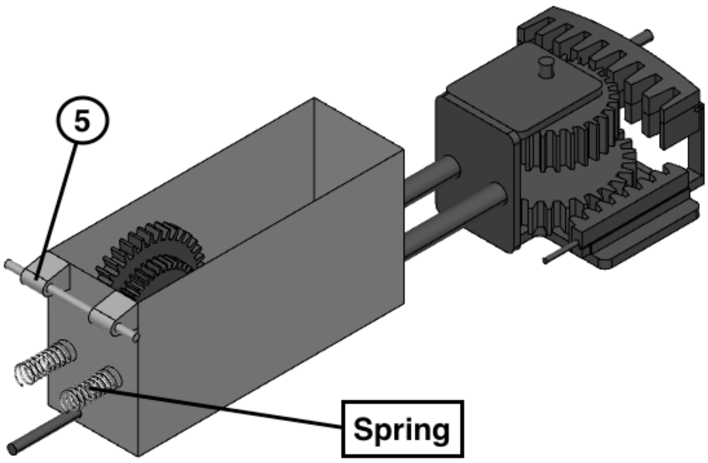


Figure 4.6: Actuation System - Solution 3

The third solution developed, is shown in fig. 4.6. As for the previous, rack-pinion are exploited in the power transmission from electric motors to the gearshift system. To perform shift a double rack-pinion system 1 is exploited, to multiply the stroke of the electric motor, not sufficient to cover the one of the main lever. The output of 1, is a force acting on the plate 4, which slides on rails 2. The complete shift system, is enclosed in a housing (fig. 4.6), which is able to rotate around hinges 5. The main lever rotates around a fix point (sphere, see section 2.3.1), as consequence the knob moves reproducing an arch, and its height varies in each point. The work of hinges and springs together, allows the housing to be always oriented, with the correct height of the knob during shift.

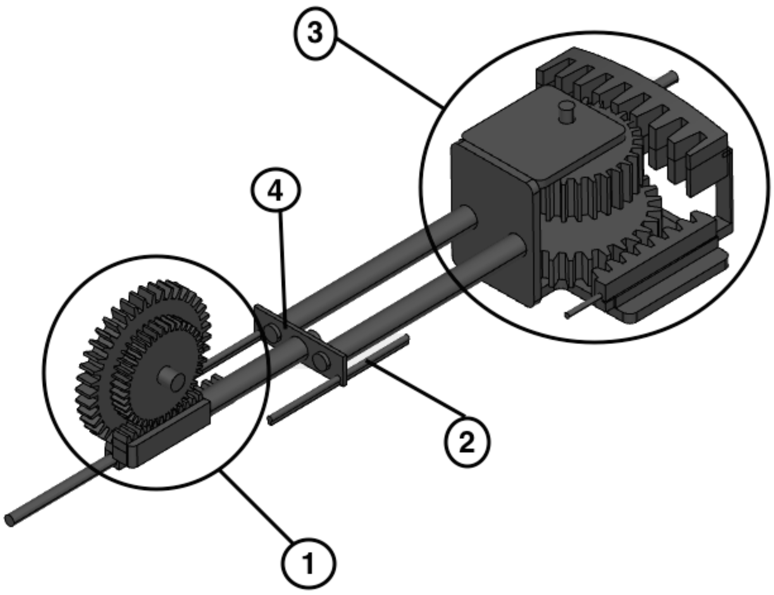


Figure 4.7: Actuation System - Solution 3 Description

The select system 3, actuated by rack-pinion 6, is a compact assembly which slides in x direction, driven by the output rack of the shift system. To perform select motion a bevel gear coupling is exploited. The pinion of system 6, engages and rotates together with bevel gear 9. The section of bevel gear 7, receives input from 9 and transmit a moment to the main lever, sliding onto a rail 8.
The complete system is adaptable to any gearshift system, simply modifying the height of the hinges, fixed on test bench, and the diameter of bevel gear 7. The advantages are: High Flexibility, Resistance to wide range of temperature, Compact (in y direction). The drawbacks:

- Inertia of the system (weight of Select System)
- Cost
- Speed achievable

The main problem in this solution is the cost, due to the number of gear exploited. The section of bevel gear 7, will require a customized manufacturing, to obtain the desired diameter and interface it with the whole system. As for the two previous concepts, the advantages are not sufficient, compared to the drawbacks, to accept this solution.
The final solution is presented in next section 4.4.

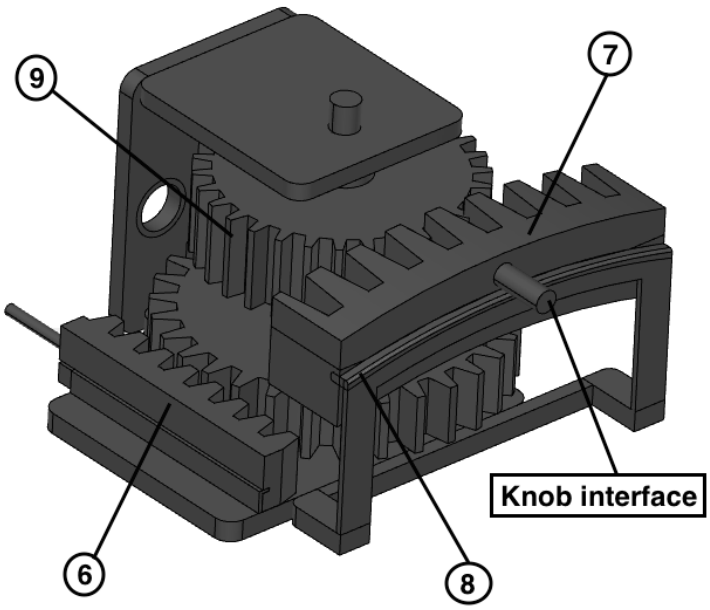


Figure 4.8: Solution 3 - Select System

4.4 Actuation System - Solution 4

The limits of the first solutions, are mainly linked to the use of teethed arches. This technical choice increases the cost of manufacturing, and requires mechanical interfaces, which affect the weight and the size of the complete system. Still the electric motors, applied as force-stroke input, together with company cables, result a good compromise to decrease the cost of the overall system.
To satisfy the requirements, exposed in section 4.1, and obtain a circular motion of the Main Lever, is necessary to re-design the concept of the arch. To avoid gears solution, is exploited an arch with a pulley rolling on it.
Notice that, the Reference System adopted in the CAD environment, is the same used for gearshift system, figure 4.9.

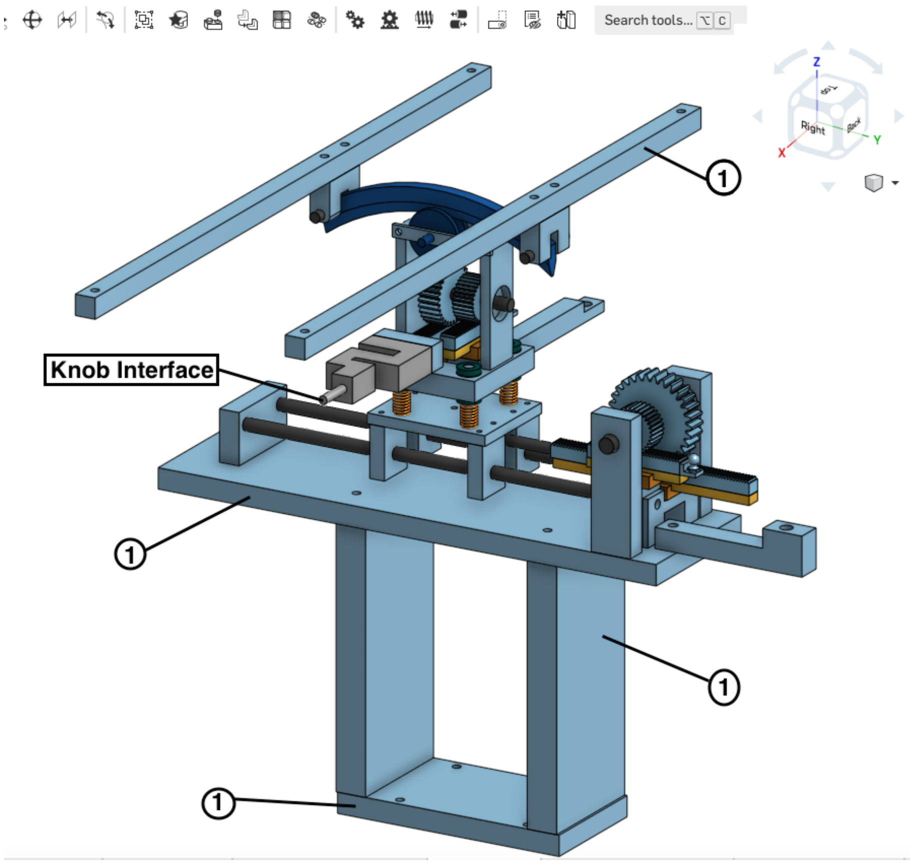


Figure 4.9: Actuation System - Solution 4

In figure 4.9, all the components marked 1, are exploited and designed to position the actuation system on the bench test, and fit it with the existing layout of YBA gearshift system.
The principle of functioning is shown in fig. 4.10. The electric motor together with a push-pull cable, is the force-stroke input acting on rack-pinion 6. The pinion of 6 is fitted on shaft with the pinion of rack-pinion system 5. Designing correctly the two pinions a multiplication of the stroke is produced on rack of 5. Through the shaft 16 the select and shift system are linked, the sliding of rack 5 along y direction, produces the motion of the complete shift system along the same direction.

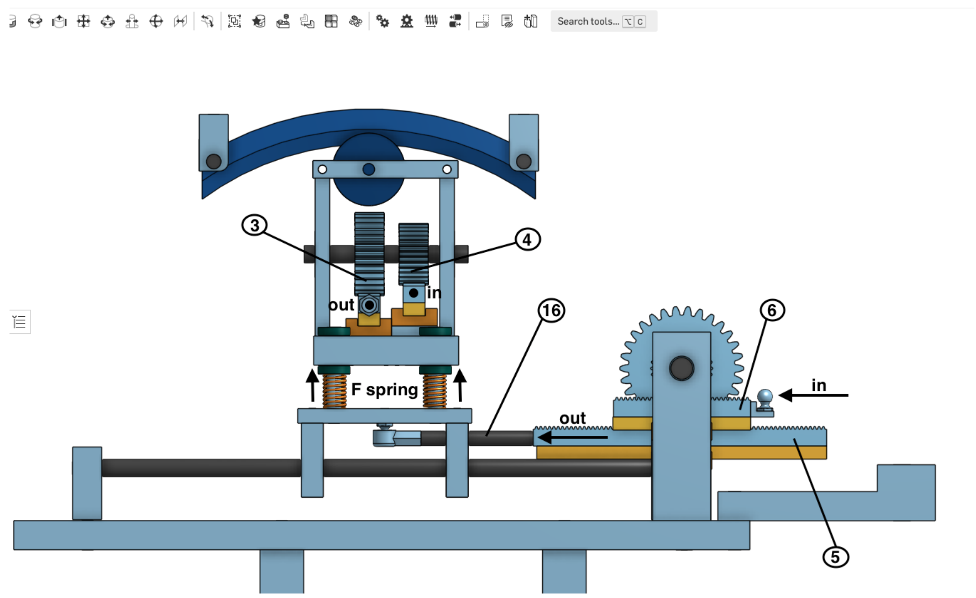


Figure 4.10: Solution 4 - Functioning Principle

The plate 14 of the shift system, slides on shafts 9. The plate 11 is the linking component between the pulley 10 and the springs 8, which allows the motion of the shift system around arch's fulcrum.
On corners of plate 11 are positioned four linear bearings 7, on plate 14 are screwed four shafts. Those shafts slide inside the bearings and allow the relative motion of 11 respect to 14, maintaining those aligned in any condition. The compression springs 8 maintain the pulley 10 always in contact along the arch 2. Springs and pulley together, allow the knob interface to be always at the correct height, and so the main lever, to perform the desired motion, an arch.
As for select system, to exploit electric motor and cable actuation, along x direction, a double rack-pinion system is adopted. The input is on rack-pinion 4 and a multiplication of the stroke is produced at the output, rack-pinion 3 (fig. 4.11). Since the tests performed on gearshift system have always a requirements concerning the force, a bi-axial load cell is positioned between rack 3 and knob interface. Through the bi-axial load cell, is possible get a feedback both in x and y direction.
Components 13 and 12, works as support for pinions shafts, both designed with seats for rotative bearings, to guarantee higher efficiency, lower internal frictions and longer service life of the system.

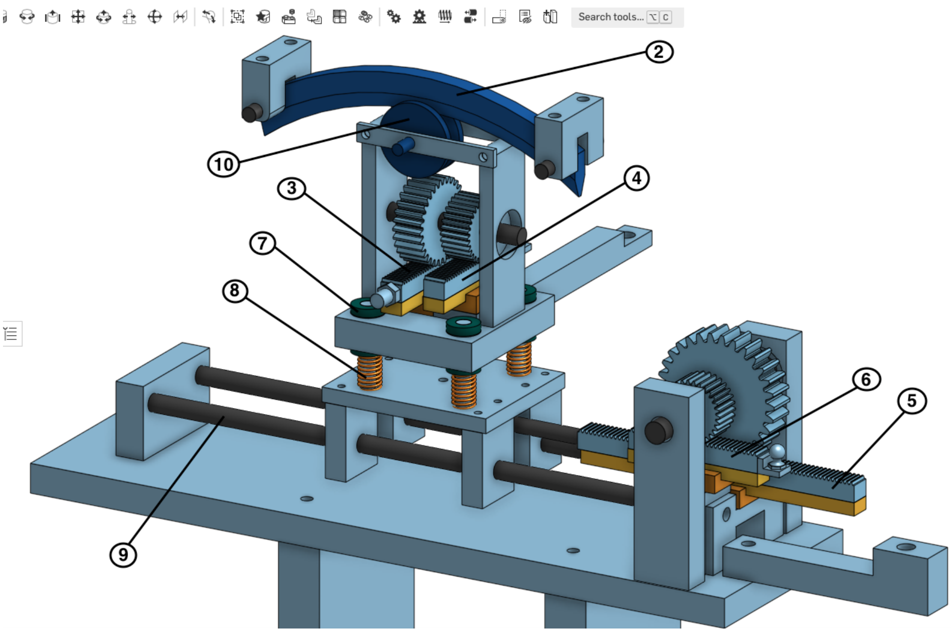


Figure 4.11: Solution 4 - Detail

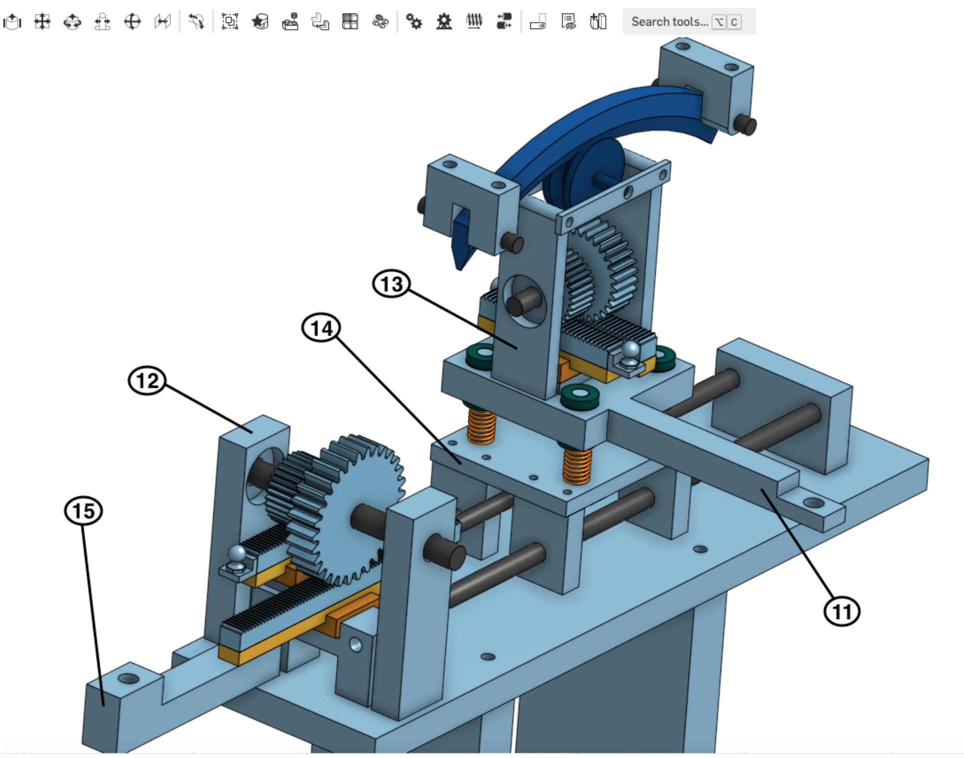


Figure 4.12: Solution 4 - Detail

The advantages of the system:

- Workstation and Electric motors: already exploited on actual systems, represent a good compromise to avoid the cost of knew actuation system, workstation and software to control it. This choice has a huge impact on the cost of the overall solution.
- Force: rack-pinion coupling allows to reach high forces at knob interface, depending on the dimensioning, and to satisfy customer specifications.
- Arch: this solution allow to obtain a force applied on the main lever applied along an arch, and satisfy the requirements of customers. Meantime, is possible to substitute the arch, with a linear guide for the pulley, to get a linear motion. As alternative, plate 11 can be positioned to a fix height respect to plate 14, obtaining the same result of linear application of force.
- Flexibility: the springs-pulley coupling makes the actuation system adaptable to any gearshift system. It is sufficient to adapt the system on the test bench and to substitute the arch. The design of the arch, based on the length of the main lever, allows to exploit the same actuation system for multiple solutions (see section 4.5).
- Compact: the dimensions of the actuation system, not considering the components marked 1, are 280 mm (z direction) and 440 mm (y direction). The reduced size, allows to improve the packaging of the test bench, and to exploit a single structure (1300x700x800 mm) to test two system in parallel. This advantage, reflects positively, on the cost (one structure required instead of two) and on the timing of tests (two systems tested contemporary).
- Resistance to Temperature: this solution doesn't presents problems for what concern the temperature. The correct choice of bearings and materials (see section 4.5), allows the actuation system to work in the range of temperature required by the customer.
- Cost: the number of components required in the system, increases the initial cost of implementation. Nevertheless, as explained previously, the manufacturing of the arch, is the only activity necessary to adapt a new gearshift system. The adoption of electric motor and workstation already present, and the compact size, are all factors useful to decrease the overall cost.
- Efficiency: the rack-pinion efficiency is typically high ($98 \div 99.5$ % [1]). The motion on y direction which exploits shafts 9 (figure 4.11), reduces the efficiency in select motion. Actually, the final solution to be adopted, is explained in section 4.5; an high efficiency sliding method is exploited.

The drawbacks of the system:

- Inertia: to perform the select motion, is required the sliding of the shift actuation system. The number of components, increases the inertia of the system. To limit the problem is fundamental the right choice of materials adopted.

4.5 Solution 4 Design

The design of chosen solution, start from the drawing of the gearshift system YBA Suzuki. The step followed are here listed:

- Evaluation and Calculation of Knob stroke.
- Definition of Neutral position in reference system
- Dimensioning of rack-pinion systems.
- Positioning of rack-pinion systems.
- Dimensioning of Linear Guides for rack.
- Dimensioning of Rotative and Linear Bearings.
- Dimensioning of Springs.
- Design of components.
- Adaptation of Actuation system to Test Bench layout.

4.5.1 Evaluation and Calculation of Knob stroke Definition of N position in reference system

To obtain all the useful parameters, concerning the gearshift system, is necessary use the drawing. The left view and the front one are reported in figure 4.13 and 4.14. On it are reported the strokes of the main lever and its length, those are the two parameters used to start the calculations.

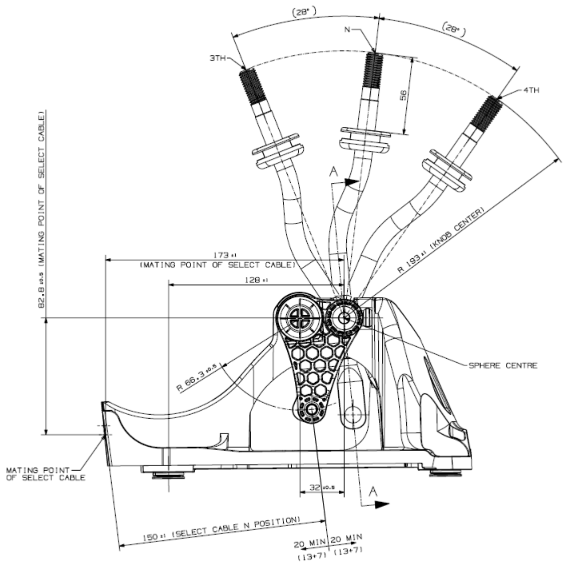


Figure 4.13: Suzuki YBA Gearshift Drawing - Left View

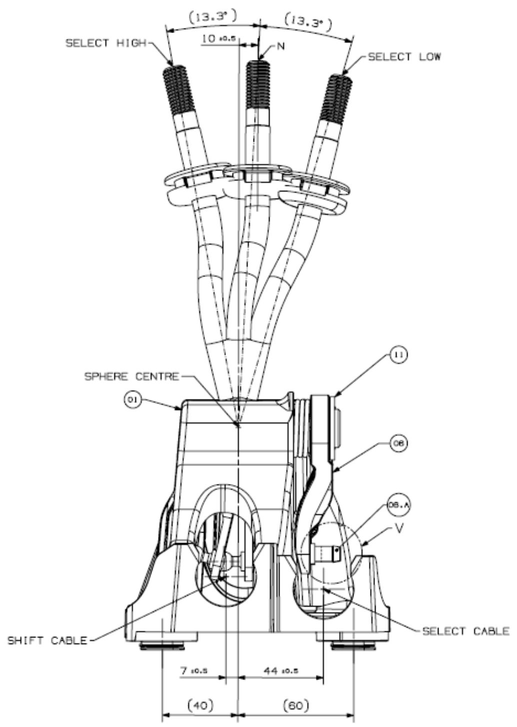


Figure 4.14: Suzuki YBA Gearshift Drawing - Front View

From the drawing is possible to define θ_L as the shift total angular stroke, θ_F as the select total angular stroke, R as the length of the main lever (from sphere to knob center). The values are reported in following table.

Table 4.1: Suzuki YBA Gearshift Drawing - Parameters

θ_L	$56°$
θ_F	$26.6°$
R	$193\ mm$

To proceed in the calculations, is useful to define the linear strokes of the arches at the knob, so to calculate the chord l.

$$l = 2R \sin\left(\frac{\theta}{2}\right)$$

The results are here reported.

Table 4.2: Suzuki YBA Gearshift Drawing - Parameters

l_L	$181.2\,mm$
l_F	$88.8\,mm$

The electric motors, force-stroke input, have a maximum stroke $l_{emMAX} = 96\ mm$, for this reason is requested a multiplication of the stroke, in shift direction, adopting a double rack-pinion system. On select direction the multiplication is not required, but is preferable not work close to the maximum stroke of the e.m., because it will influence the speed and frequency achievable. Moreover a requirements for the actuation system is flexibility (see section 3), so a double rack-pinion system is adopted even in select direction, to cover wider angular displacement of main lever, on other systems. The multiplication ratio can be easily defined as

$$r = \left(\frac{l_i}{l_{emMAX}}\right)$$

The multiplication ratio required in shift direction is $r_L = 1.88$. To avoid e.m. working at l_{emMAX} is chosen $r_L = 2$. In select direction is chosen an arbitrary ratio $r_F = 1.35$, for reasons exposed above.

4.5.2 Dimensioning of rack-pinion systems Positioning of rack-pinion systems

Before the dimensioning of the rack-pinion systems, is useful recall some terminology and formula concerning gears ([1] [2] [3]).

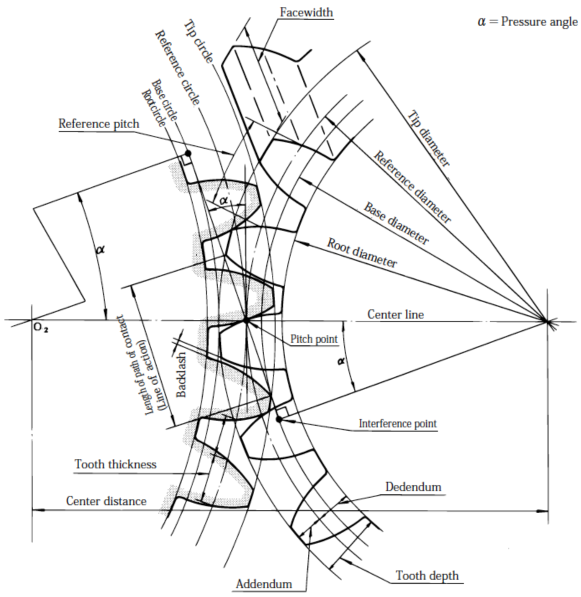


Figure 4.15: Spur Gear meshing

The *module* is the unit of size that indicates how big or small a gear is. Is the ratio of the reference diameter of the gear divided by the number of teeth.[2]

$$m = \frac{d}{z}$$

The module typically is chosen between normalized values, those are the most common produced and adopted in mechanical applications, easily available.
The contact point of the two involute, in figure 4.16, slides along the common tangent of the two base circle as the rotation occurs. The common tangent is called the line of contact, or the *line of action*.[1]

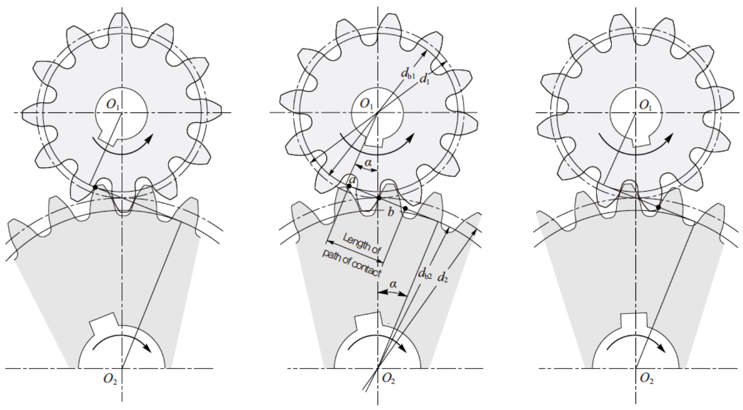


Figure 4.16: Spur Gear meshing

The intersection between the line of action and the center line is defined as the *pitch point* (fig. 4.15)
The *reference pitch* $p = \pi m$.
The *pressure angle* is defined as the angle between the tangent to both *renference circles* and the *line of action*. A pair of gear can only mesh correctly if the pitches and the pressure angle are the same.[1] The rack is assumed as a spur gear with infinite radius.
To satisfy the multiplication ratio, calculated previously, the following equation is exploited, which express the relation between linear displacement of the rack and angular displacement of the pinion.

$$l = \left(\frac{z\delta}{360}\right)\pi m$$

The following parameter and procedure are exploited, to define the number of teeth of the pinions.

Table 4.3: Rack-pinion Calculation - Parameters

δ_{in}	Angular displacement of Input pinion
δ_{out}	Angular displacement of Output pinion
z_{in}	Teeth of Input pinion
z_{out}	Teeth of Output pinion
l_{in}	Linear displacement of Input rack
l_{out}	Linear displacement of Output rack

The pinions are integral with the shaft, so angular displacement $\delta_{in} = \delta_{out}$. To satisfy the multiplication ratio $l_{out} = r_i l_{in}$, here is reported the procedure.

$$l_{in} = \left(\frac{z_{in}\delta_{in}}{360}\right)\pi m \qquad l_{out} = \left(\frac{z_{out}\delta_{out}}{360}\right)\pi m \qquad \left(\frac{z_{out}\delta_{out}}{360}\right)\pi m = r_i\left(\frac{z_{in}\delta_{in}}{360}\right)\pi m$$

After simplification

$$z_{out} = r_i z_{in}$$

Can be noticed that the relation linking the racks, is directly reflected in number of teeth of the pinions, as they are integral.
When the number of gear teeth to be cut becomes small, the generating tool will sweep out its path, removing some of the profile, and producing an undercut tooth form (fig. 4.17). To prevent undercut, some correction must be introduced: minimum number of teeth or profile shifting.[2]

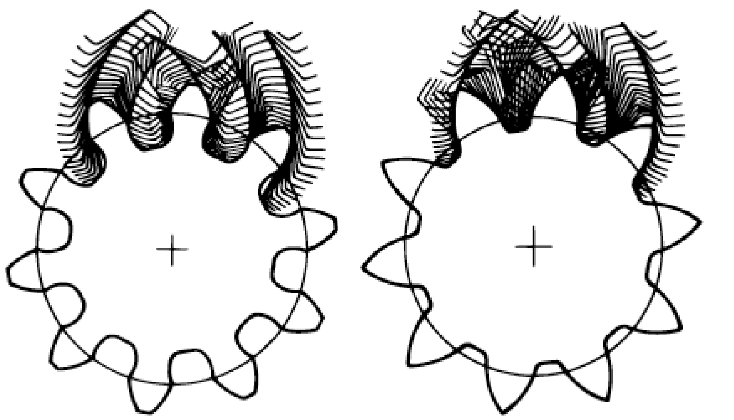


Figure 4.17: Spur Gear Profile Shifting

The minimum number of teeth can be calculated as follow

$$z_{min} = \frac{2}{\sin^2\alpha}$$

With $\alpha = 20°$ results to be $z_{min} = 17$.
The advantages introduced by profile shifting are: Prevent undercut with small number of teeth, Adjust center distance. Is possible define a profile shift coefficient without undercut x_s.

$$x_s = 1 - \left(\frac{z}{2}\right)\sin^2\alpha$$

The coupling of a rack-pinion system, with profile shifting, is exposed in figure 4.18.

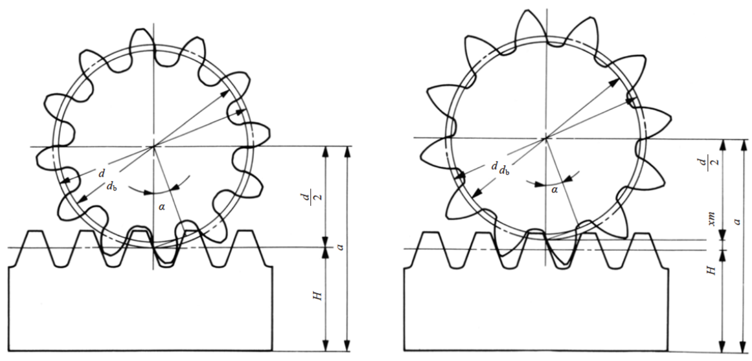


Figure 4.18: Rack-pinion Profile Shifting

To select spur gears and racks, is used Chiaravalli's catalogue[4], as it is a reference manufacturer in mechanical components. Moreover the adoption of normalized components, avoids the costs of special manufacturing. To limit undercut the choice is oriented toward gears with $z > z_{min}$, respecting also the requirements $z_{out} = r_i z_{in}$. Chiaravalli doesn't apply a profile shifting, for gear correction.
The module is chosen arbitrary between the normalized as $m = 2$ for all the components. In the following table are reported the parameters of gears chosen.

Table 4.4: Rack-pinion Calculation - Parameters

Component	z	d_r	d_t	Material
Input Gear Shift	20	40 mm	44 mm	C45 UNI 7845
Output Gear Shift	40	80 mm	84 mm	C45 UNI 7845
Input Gear Select	20	40 mm	44 mm	C45 UNI 7845
Output Gear Select	27	54 mm	58 mm	C45 UNI 7845
Component	H	H_t	B	Material
Rack Shift	18 mm	20 mm	20 mm	C40
Rack Select	18 mm	20 mm	20 mm	C40

The next step is the calculation of dimensions of a spur gear-rack system, in addition to the one furnished by the catalogue, to proceed then with Bending Strength and Surface Durability validation. Here are reported in sequence the calculations performed.
Root Diameter $d_{rt} = d_r - 2.25m$
Base Diameter $d_b = d_r \cos\alpha$
Addendum $h_a = m$
Dedendum $h_d = 1.25m$
Tooth depth $h = 2.25m$

Mounting Distance $a = \frac{zm}{2} + H$

Table 4.5: Rack-Pinion Dimensions - Shift System

Item	Symbol	Input Pinion	Input Rack	Output Pinion	Output Rack
Module	m	2			
Pressure angle $[deg]$	α	20°			
Number of teeth	z	20	—	40	—
Pitch line $[mm]$	H	—	18	—	18
Reference diameter $[mm]$	d_r	40	—	80	—
Mounting distance $[mm]$	a	38		58	
Addendum $[mm]$	h_a	2			
Dedendum $[mm]$	h_d	2.5			

Table 4.6: Rack-Pinion Dimensions - Select System

Item	Symbol	Input Pinion	Input Rack	Output Pinion	Output Rack
Module	m	2			
Pressure angle $[deg]$	α	20°			
Number of teeth	z	20	—	27	—
Pitch line $[mm]$	H	—	18	—	18
Reference diameter $[mm]$	d_r	40	—	54	—
Mounting distance $[mm]$	a	38		45	
Addendum $[mm]$	h_a	2			
Dedendum $[mm]$	h_d	2.5			

The validation of the gears, is performed considering: Bending Strength and Surface Durability. In this section are reported the calculations' process and the results. The formulas are based on JGMA (Japan Gear Manufacturers Association's Standards), in particular:

- JGMA 401-01 Bending Strength Formulas for Spur Gears and Helical Gears.
- JGMA 402-01 Surface Durability Formulas for Spur Gear and Helical Gear.

Bending Strength Validation

The calculation concerning the Bending Strength, starts from evaluation of tangential force F_t, power P and torque T, all calculated at the *pitch diameter* d_w. In rack-pinion coupling the *pitch diameter* is equal to the *reference diameter* d_r and *pressure angle* α is equal to *working pressure angle* α_w.

$$d_w = \frac{d_b}{\cos\alpha_w} \qquad d_b = d_r\cos\alpha \qquad d_r = d_w = zm$$

Here are reported the formulas to proceed in Bending Strength evaluation.

$$F_t = \frac{1000P}{v} = \frac{19614T}{d_b} \tag{4.1}$$

$$P = \frac{F_t v}{1000} \tag{4.2}$$

$$T = \frac{F_t d_b}{19614} = \frac{9552P}{n} \tag{4.3}$$

The tangential speed is calculated as $v = \frac{d_b n}{19100}$ $[m/s]$.
To satisfy the Bending Strength, the tangential force must not exceed the allowable one at *reference diameter* $F_{t,lim}$, calculated considering bending stress at the root of the tooth.

$$F_t \leq F_{t,lim} \tag{4.4}$$

Equally the condition must be respected, when considering the bending stress at the root σ_f, calculated on the basis of tangential force at the *reference diameter*.

$$\sigma_f \leq \sigma_{f,lim} \tag{4.5}$$

The calculation of $F_{t,lim}$ can be performed with equation 4.6.

$$F_{t,lim} = \sigma_{f,lim}\frac{mb}{Y_F Y_\epsilon Y_\beta}\left(\frac{K_L K_{FX}}{K_V K_O}\right)\frac{1}{S_F} \tag{4.6}$$

Equation 4.6 can be easily converted into stress as follow.

$$\sigma_f = F_t\frac{Y_F Y_\epsilon Y_\beta}{mb}\left(\frac{K_V K_O}{K_L K_{FX}}\right)S_F \tag{4.7}$$

After the presentation of the main equations, are explained the variables inside eq. 4.6. Each of these variables is directly linked to the gears characteristics, in this case to the rack-pinion coupling.

- Facewidth b [mm]
  In case the gears coupled have different facewidth, consider the wider b_w and the narrower b_n. The possible conditions are the following:
  $$\begin{cases} b_w - b_n \leq m & \text{The facewidth values can be used in equation 4.6.} \\ b_w - b_n \geq m & \text{The wider values would be changed to } b_n + m\text{, while} \\ & \quad\text{the narrower would be unchanged.} \end{cases}$$

- Tooth Profile Factor Y_F

  The Tooth Profile Factor Y_F, can be evaluated from the graph in figure 4.19. The graph is based on following parameters:

  – Equivalent number of teeth $z_v = \frac{z}{\cos^3 \beta}$.

  – Profile Shift Coefficient x.

  – Theoretical Undercut Limit.

  – Narrow Tooth Top Limit.

  The graph is specific for angle pressure $\alpha = 20°$, which is actually the angle of chosen gears.

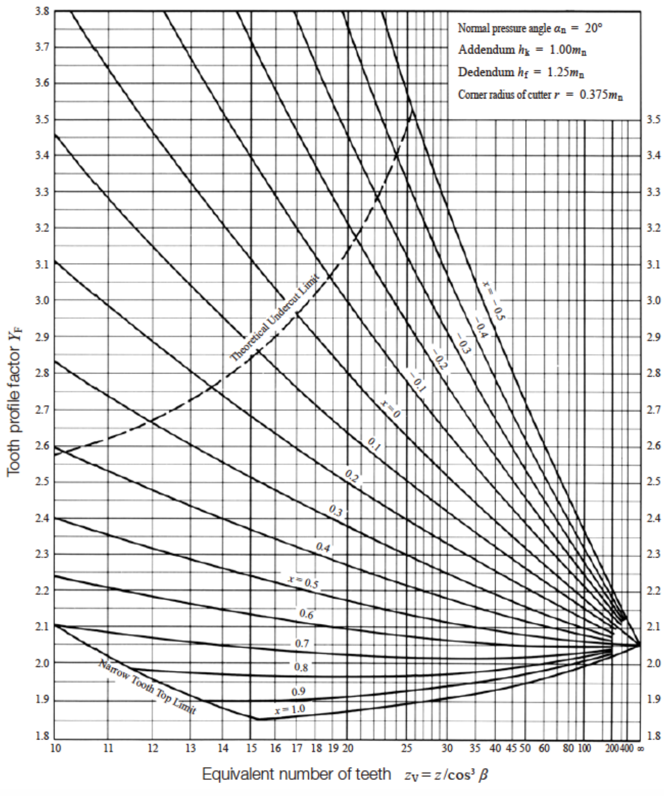


Figure 4.19: Profile Factor Y_F [1]

The values of Tooth Profile Factor, introduced in equations, are reported in table 4.7 for each component.

Table 4.7: Tooth Profile Factor Y_F

Item	Pinion	Rack
Shift Input	2.79	2.066
Shift Output	2.4	2.066
Select Input	2.79	2.066
Select Output	2.58	2.066

- Load Sharing Factor Y_ϵ

  The Load Sharing Factor Y_ϵ, is the reciprocal of the transverse contact ratio ϵ_α.

$$Y_\epsilon = \frac{1}{\epsilon_\alpha} \tag{4.8}$$

Considering a spur gear, the parameter, can be calculated with the following equation

$$\epsilon_\alpha = \frac{\sqrt{r_{t1}^2 - r_{r1}^2} + \sqrt{r_{t2}^2 - r_{r2}^2} - a \sin \alpha_w}{\pi m \cos \alpha} \tag{4.9}$$

In figure 4.20, is reported a table for the evaluation of the Transverse Contact Ratio. The table reports values for pressure angle $\alpha = 20°$. For the aim of this project, will be considered, on the table, the coordinate *"Rack"* and the corresponding number of teeth of the pinion.

$(\alpha_0 = 20°)$

No. of teeth	17	20	25	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	110	120
17	1.514																			
20	1.535	1.557																		
25	1.563	1.584	1.612																	
30	1.584	1.605	1.633	1.654																
35	1.603	1.622	1.649	1.670	1.687															
40	1.614	1.635	1.663	1.684	1.700	1.714														
45	1.625	1.646	1.674	1.695	1.711	1.725	1.736													
50	1.634	1.656	1.683	1.704	1.721	1.734	1.745	1.755												
55	1.642	1.664	1.691	1.712	1.729	1.742	1.753	1.763	1.771											
60	1.649	1.671	1.698	1.719	1.736	1.749	1.760	1.770	1.778	1.785										
65	1.655	1.677	1.704	1.725	1.742	1.755	1.766	1.776	1.784	1.791	1.797									
70	1.661	1.682	1.710	1.731	1.747	1.761	1.772	1.781	1.789	1.796	1.802	1.808								
75	1.666	1.687	1.714	1.735	1.752	1.765	1.777	1.786	1.794	1.801	1.807	1.812	1.817							
80	1.670	1.691	1.719	1.740	1.756	1.770	1.781	1.790	1.798	1.805	1.811	1.817	1.821	1.826						
85	1.674	1.695	1.723	1.743	1.760	1.773	1.785	1.794	1.802	1.809	1.815	1.821	1.825	1.830	1.833					
90	1.677	1.699	1.726	1.747	1.764	1.777	1.788	1.798	1.806	1.813	1.819	1.824	1.829	1.833	1.837	1.840				
95	1.681	1.702	1.729	1.750	1.767	1.780	1.791	1.801	1.809	1.816	1.822	1.827	1.832	1.836	1.840	1.844	1.847			
100	1.683	1.705	1.732	1.753	1.770	1.783	1.794	1.804	1.812	1.819	1.825	1.830	1.835	1.839	1.843	1.846	1.850	1.853		
110	1.688	1.710	1.737	1.758	1.775	1.788	1.799	1.809	1.817	1.824	1.830	1.835	1.840	1.844	1.848	1.852	1.855	1.858	1.863	
120	1.693	1.714	1.742	1.762	1.779	1.792	1.804	1.813	1.821	1.828	1.834	1.840	1.844	1.849	1.852	1.856	1.859	1.862	1.867	1.871
RACK	1.748	1.769	1.797	1.817	1.834	1.847	1.859	1.868	1.876	1.883	1.889	1.894	1.899	1.903	1.907	1.911	1.914	1.917	1.922	1.926

$$\varepsilon_\alpha = \frac{\sqrt{r_{k1}^2 - r_{g1}^2} + \sqrt{r_{k2}^2 - r_{g2}^2} - a \sin \alpha_b}{\pi m \cos \alpha_0}$$

Figure 4.20: Transverse Contact Ratio ϵ_α [1]

The selected values of Load Sharing Factor Y_ϵ, function of Transverse Contact Ratio ϵ_α, are reported in table 4.8.

Table 4.8: Load Sharing Factor Factor Y_ϵ

Item	Pinion & Rack
Shift Input	0.565
Shift Output	0.541
Select Input	0.565
Select Output	0.554

- Helix Angle Factor Y_β
  The Helix Angle Factor Y_β is defined as follow

$$\begin{cases} 0 \leq \beta \leq 30° & Y_\beta = 1 - \frac{\beta}{120} \\ \beta \geq 30° & Y_\beta = 0.75 \end{cases} \tag{4.10}$$

  The Y_β factor, accounts the effects of helical teeth of the contact and force exchange. The design difference between spur and helical gear, has a direct influence on the bending strength. The studied rack-pinion systems exploits spur gear, so from eq. 4.10 can be defined a factor $Y_\beta = 1$.

- Life Factor K_L
  The Life Factor K_L introduces the influence of forecast cyclic repetitions in components lifetime, and accounts the total load which undergoes the gear. The value of K_L can be defined from table in figure 4.32, as function of cyclic repetitions.

No. of cyclic repetitions	Hardness (1) H_B120 ~ 220	Hardness (2) H_B221 or over	Gears w. carburizing/nitriding
10000 or fewer	1.4	1.5	1.5
Approx. 100000	1.2	1.4	1.5
Approx. 10^6	1.1	1.1	1.1
10^7 or greater	1.0	1.0	1.0

Figure 4.21: Life Factor K_L [1]

In section 3, are reported the main features required by the new actuation System. Reliability and robustness are two objectives, as consequence in the design of rack-pinion system will be considered $K_L = 1$. The aim of this choice is to design a system, which guarantees long lasting activity (over 10^7 cyclic repetitions).

- Size Factor of Root Stress K_{FX}
  Generally the Size Factor of Root Stress is assumed $K_{FX} = 1$.

- Dynamic Load Factor K_V
  The Dynamic Load Factor is a parameter function of tooth profile precision and tangential speed at *reference diameter*. The value of K_V can be defined by table shown in figure

Precision grade of gears from JIS B 1702		Tangential speed at working pitch circle (m/s)						
Tooth profile								
Unmodified	Modified	1 or under	Over 1 to 3 incl.	Over 3 to 5 incl.	Over 5 to 8 incl.	Over 8 to 12 incl.	Over 12 to 18 incl.	Over 18 to 25 incl.
	1	—	—	1.0	1.0	1.1	1.2	1.3
1	2	—	1.0	1.05	1.1	1.2	1.3	1.5
2	3	1.0	1.1	1.15	1.2	1.3	1.5	
3	4	1.0	1.2	1.3	1.4	1.5		
4	—	1.0	1.3	1.4	1.5			
5	—	1.1	1.4	1.5				
6	—	1.2	1.5					

Figure 4.22: Load Factor K_V [1]

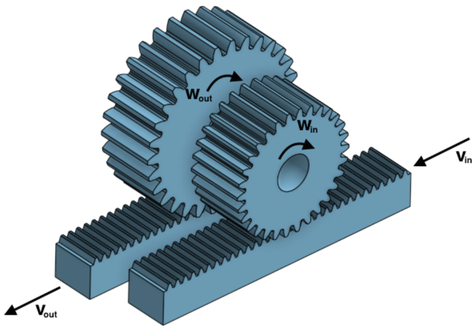
The evaluation of the tangential speed, is performed with the following procedure.

· Conditions:

- Cycle Frequency 1 Hz.
- Cycle Stroke assumed equal to complete stroke (both in Shift l_L and Select l_F).
- Linear speed of rack assumed equal to tangential speed of pinion at reference diameter.

· Calculations:

Figure 4.23: Rack-Pinion Speed



Shift Rack-Pinion speeds:

$$v_{out} = \frac{l_L}{frequency} = 181.2\, mm/s \qquad \omega_{in} = \frac{v_{in}}{r_{in}} = \omega_{out} = \frac{v_{out}}{r_{out}} = 4.53\, rad/s = 43.26\, rpm$$

To maintain a safety range in calculations, is chosen $\omega_{in} = \omega_{out} = 50\ rpm$ to perform calculations on Shift system.

Select Rack-Pinion speeds:

$$v_{out} = \frac{l_F}{frequency} = 88.8\, mm/s \qquad \omega_{in} = \frac{v_{in}}{r_{in}} = \omega_{out} = \frac{v_{out}}{r_{out}} = 3.28\, rad/s = 31.4\, rpm$$

To maintain a safety range in calculations, is chosen $\omega_{in} = \omega_{out} = 40\ rpm$ to perform calculations on Select system.

- Overload Factor K_O
  The Overload Factor introduces, in bending strength equation, the contribution of an overload, applied on gears' teeth, respect to the nominal value expected. Can be calculated as the ratio between actual and nominal force, with equation 4.11.

$$K_O = \frac{F_{t,act}}{F_{t,nom}} \tag{4.11}$$

If tangential force F_t is unknown, the Overload Factor can be evaluated through table in figure 4.24 . Coupling load grade is the guiding parameter.

Impact from Prime Mover	Impact from Load Side of Machine		
	Uniform Load	Medium Impact Load	Heavy Impact Load
Uniform Load (Motor, Turbine, Hydraulic Motor)	1.0	1.25	1.75
Light Impact Load (Multicylinder Engine)	1.25	1.5	2.0
Medium Impact Load (Single Cylinder Engine)	1.5	1.75	2.25

Figure 4.24: Overload Factor K_O [1]

In this project of thesis is assumed a *good tooth contact*, as consequence the load transmission will be uniform, without impact between teeth. The Overload Factor is assumed $K_O = 1$.

- Safety Factor for Bending Failure S_F
  The Safety Factor for Bending Failure is quite a complicated issue to be determined. Typically this parameter is set $S_F = 1.2$ at least, to maintain a safety range in design, in this project of thesis the Safety Factor is assumed $S_F = 1.3$.

- Allowable Bending Stress at the Root $\sigma_{f,lim}$
  The Allowable Bending Stress at the Root can be evaluated with the tables reported in following figures. In these tables $\sigma_{f,lim}$ is the quotient of fatigue limit, under pulsating tension, divided by the stress concentration factor. The values reported are valid for unidirectional loaded gear, in bidirectional load, the value of allowable bending stress is 2/3 of the ones reported.
  The evaluation of $\sigma_{f,lim}$ is deeply influenced by the material used for rack-pinion system

Table 10.5 Gears without surface hardening

Material (Arrows indicate the ranges)		Hardness H_B	Hardness H_V	Tensile strength lower limit kgf/mm² (Reference)	σ_{Flim} kgf/mm²
Cast steel gear	SC37			37	10.4
	SC42			42	12
	SC46			46	13.2
	SC49			49	14.2
	SCC3			55	15.8
				60	17.2
Normalizing carbon steel gear		120	126	39	13.8
		130	136	42	14.8
		140	147	45	15.8
	S25C	150	157	48	16.8
		160	167	51	17.6
		170	178	55	18.4
	S35C	180	189	58	19
		190	200	61	19.5
	S43C	200	210	64	20
	S48C	210	221	68	20.5
	S53C	220	231	71	21
	S58C	230	242	74	21.5
		240	252	77	22
		250	263	81	22.5
Quenched and tempered carbon steel gear		160	167	51	18.2
		170	178	55	19.4
		180	189	58	20.2
		190	200	61	21
	S35C	200	210	64	22
		210	221	68	23
		220	231	71	23.5
		230	242	74	24
	S43C, S48C	240	252	77	24.5
		250	263	81	25
	S53C	260	273	84	25.5
	S58C	270	284	87	26
		280	295	90	26
		290	305	93	26.5
Quenched and tempered alloy steel gear		220	231	71	25
		230	242	74	26
		240	252	77	27.5
		250	263	81	28.5
	SMn443	260	273	84	29.5
		270	284	87	31
		280	295	90	32
	SNC836	290	305	93	33
	SCM435	300	316	97	34
	SCM440	310	327	100	35
	SNCM439	320	337	103	36.5
		330	347	106	37.5
		340	358	110	39
		350	369	113	40
		360	380	117	41

Figure 4.25: $\sigma_{f,lim}$ Gear without Surface Hardening [1]

Table 10.6 Induction hardened gears

		Material (Arrows indicate the ranges)	Heat treatment before induction hardening	Core hardness H_B	Core hardness H_V	Surface Hardness (1) H_V	σ_{Flim} kgf/mm²
Hardened throughout	Structural carbon steel	S48C, S43C	Normalized	160	167	More than 550	21
				180	189	〃	21
				220	231	〃	21.5
				240	252	〃	22
		S48C, S43C	Quenched and tempered	200	210	More than 550	23
				210	221	〃	23.5
				220	231	〃	24
				230	242	〃	24.5
				240	252	〃	25
				250	263	〃	25
	Structural alloy steel	SMn443, SCM440, SNCM439, SCM435, SNC836	Quenched and tempered	230	242	More than 550	27
				240	252	〃	28
				250	263	〃	29
				260	273	〃	30
				270	284	〃	31
				280	295	〃	32
				290	305	〃	33
				300	316	〃	34
				310	327	〃	35
				320	337	〃	36.5
Hardened except root area							75% of the above

Figure 4.26: $\sigma_{f,lim}$ Induction Hardened Gear [1]

Table 10.7 Carburized and quenched gears

	Material (Arrows indicate the ranges)	Core hardness H_B	Core hardness H_V	σ_{Flim} kgf/mm²
Structural carbon steel	S15C, S15CK	140	147	18.2
		150	157	19.6
		160	167	21
		170	178	22
		180	189	23
		190	200	24
Structural alloy steel	SCM415, SCM420, SNC415, SNCM420, SNC815	220	231	34
		230	242	36
		240	252	38
		250	263	39
		260	273	41
		270	284	42.5
		280	295	44
		290	305	45
		300	316	46
		310	327	47
		320	337	48
		330	347	49
		340	358	50
		350	369	51
		360	380	51.5
		370	390	52

Figure 4.27: $\sigma_{f,lim}$ Carburized and Quenched Gear [1]

The components, selected from Chiaravalli's catalogue [4], are made of C45 UNI7845, a medium carbon steel largely exploited in automotive manufacturing. The main properties of this material are reported here, from [5].

Quenched and tempered, +QT:

Diameter d [mm]	< 16	>16 – 40	>40 – 100	>100 – 160	>160 – 250
Thickness t [mm]	< 8	8<t<20	20<t<60	60<t<100	100<t<160
0,2% proof stress $R_{p0,2}$ [N/mm²]	min. 490	min. 430	min. 370	-	-
Tensile strength R_m [N/mm²]	700 - 850	650 - 800	630 - 780	-	-
Fracture elongation A_5 [%]	min. 14	min. 16	min. 17	-	-
Reduction of area Z [%]	min. 35	min. 40	min. 45	-	-

Figure 4.28: C45 UNI7845 - Quenched and Tempered

Normalised, +N:

Diameter d [mm]	< 16	>16 – 100	>100 – 250
Thickness t [mm]	< 16	16<t<100	100<t<250
0,2% proof stress $R_{p0,2}$ [N/mm²]	min. 340	min. 305	min. 275
Tensile strength R_m [N/mm²]	min. 620	min. 580	min. 560
Fracture elongation A_5 [%]	min. 14	min. 16	min. 16

Figure 4.29: C45 UNI7845 - Normalized

The evaluation of Allowable Bending Stress at the Root, produces the following results, reported in table 4.9.

Table 4.9: Allowable Bending Stress at the Root $\sigma_{f,lim}$ $[MPa]$

Item	Pinion	Rack
Shift Input	160.175	130.75
Shift Output	160.175	130.75
Select Input	160.175	130.75
Select Output	160.175	130.75

After the selection of each parameter, related to rack-pinion system exploited, the calculation to validate the components, under Bending Strength targets are performed. In the following table are reported the results.

Table 4.10: Bending Strength Validation - Rack Pinion Shift Input

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	2891	3197.7
Allowable Torque $[Nm]$	57.82	—
Allowable Power $[KW]$	0.302	0.334

Table 4.11: Bending Strength Validation - Rack Pinion Shift Output

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	3515.5	3339.4
Allowable Torque $[Nm]$	140.62	—
Allowable Power $[KW]$	0.736	0.699

Table 4.12: Bending Strength Validation - Rack Pinion Select Input

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	2891	3197.7
Allowable Torque $[Nm]$	57.82	—
Allowable Power $[KW]$	0.242	0.268

Table 4.13: Bending Strength Validation - Rack Pinion Select Output

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	3194	3263.6
Allowable Torque $[Nm]$	86.24	—
Allowable Power $[KW]$	0.361	0.369

The results of calculations shows that the rack-pinion chosen largely satisfy the requirements. The values obtained are all greater than the ones defined in section 2.3.3.

Surface Durability Validation

The process for Surface Durability Validation, is derived from KHK Gear Technical Reference [1], and follows JGMA standards. In order to satisfy the Surface Durability, are useful the equations related to the tangential force at the reference diameter F_t. Equations 4.1, 4.3 and 4.2 exposed for Bending Strength, are still valid.
In Surface Durability Validation, $F_{t,lim}$ is calculated considering the allowable Hertz

stress σ_H, to satisfy the same condition expressed previously at 4.4. The actual Hertz stress, calculated at the *reference diameter*, should respect the condition

$$\sigma_H \leq \sigma_{H,lim} \tag{4.12}$$

For the calculation of $F_{t,lim}$ the following equation can be used.

$$F_{t,lim} = \sigma_{H,lim}^2 db_H \frac{i}{i \pm 1} \left(\frac{K_{HL} Z_L Z_R Z_V Z_W K_{HX}}{Z_H Z_M Z_\epsilon Z_\beta} \right)^2 \frac{1}{K_{H\beta} K_V K_O} \frac{1}{S_H^2} \tag{4.13}$$

From eq. 4.13 is derived the equation for the calculation of the Hertz Stress.

$$\sigma_H = \sqrt{\frac{F_t}{db_H} \frac{i \pm 1}{i}} \frac{Z_H Z_M Z_\epsilon Z_\beta}{K_{HL} Z_L Z_R Z_V Z_W K_{HX}} \sqrt{K_{H\beta} K_V K_O} S_H \tag{4.14}$$

In equations 4.13 and 4.14 the factor $\frac{i}{i \pm 1}$ has:

- Symbol " + " for two external gears meshing.
- Symbol " − " for internal and external gears meshing.
- Value 1 for rack-pinion system

As for Bending Strength Validation, are explained the variables exploited in equation 4.13.

- Effective Facewidth b_H
  When gears with different facewidth mesh, is considered the thinner one as the Effective Facewidth, for the calculation of surface strength.

- Zone Factor Z_H
  The Zone Factor Z_H is defined with the following equation.

$$Z_H = \sqrt{\frac{2 \cos \beta_g \cos \alpha_{bs}}{\cos^2 \alpha_s \sin \alpha_{bs}}} \tag{4.15}$$

  Where $\beta_g = \tan^{-1}(\tan \beta \cos \alpha_s)$ is the base helix angle, α_{bs} is the working transverse pressure angle and α_s is the transverse pressure angle.
  As alternative Z_H can be evaluated through the graph reported in figure 4.30, here the parameter is function of: pressure angle $\alpha = 20°$, profile shift coefficient x_1 and x_2, number of teeth z_1 and z_2, helix angle β.

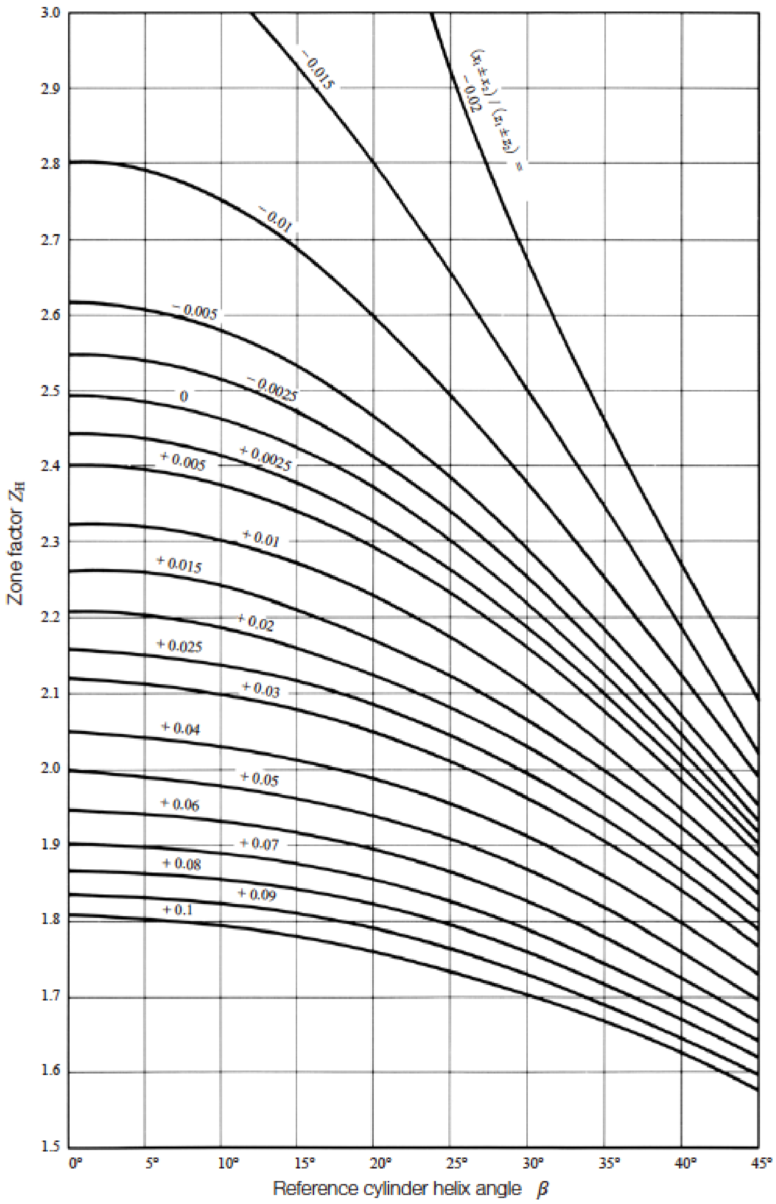


Figure 4.30: Zone Factor Z_H [1]

The symbols "+" and "−" in the graph follow the same rules of equation 4.13. In table 4.14 are reported the values of Zone Factor introduced in calculations.

Table 4.14: Zone Factor Z_H

Item	Pinion & Rack
Shift Input	2.5
Shift Output	2.5
Select Input	2.5
Select Output	2.5

- Material Factor Z_M
  The Material Factor Z_M is defined by equation

$$Z_M = \sqrt{\frac{1}{\pi \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)}} \tag{4.16}$$

  Where ν is the Poisson's ratio and E is the Young's modulus.
  The table in figure 4.31, reports several combinations of materials and their Material Factor Z_M.

Gear				Meshing gear				Material factor Z_M (kgf/mm²)$^{0.5}$
Material	Symbol	Young's modulus E kgf/mm²	Poisson's ratio ν	Material	Symbol	Young's modulus E kgf/mm²	Poisson's ratio ν	
Structural steel	* (1)	21000	0.3	Structural steel	* (1)	21000	0.3	60.6
				Cast steel	SC	20500		60.2
				Ductile cast iron	FCD	17600		57.9
				Gray cast iron	FC	12000		51.7
Cast steel	SC	20500		Cast iron	SC	20500		59.9
				Ductile cast iron	FCD	17600		57.6
				Gray cast iron	FC	12000		51.5
Ductile cast iron	FCD	17600		Ductile cast iron	FCD	17600		55.5
				Gray cast iron	FC	12000		50.0
Gray cast iron	F C	12000		Gray cast	FC	12000		45.8

Figure 4.31: Material Factor Z_M [1]

The Material Factor is defined equally for all components, as the material is the same, $Z_M = 189.783\ MPa$.

- Contact Ratio Factor Z_ϵ
  The Contact Ratio Factor is defined as follow.

$$\begin{cases} SpurGear & Z_\epsilon = 1 \\ HelicalGear\ \epsilon_\beta \leq 1 & Z_\epsilon = \sqrt{1 - \epsilon_\beta + \frac{\epsilon_\beta}{\epsilon_\alpha}} \\ HelicalGear\ \epsilon_\beta > 1 & Z_\epsilon = \sqrt{\frac{1}{\epsilon_\alpha}} \\ With\ \epsilon_\beta = \frac{b_H \sin\beta}{\pi m} \end{cases} \tag{4.17}$$

The gears adopted are of spur kind, so the Contact Ratio Factor is easily defined.

- Helix Angle Factor Z_β
  The Helix Angle Factor usually is assumed $Z_\beta = 1$.

- Life Factor K_{HL}
  The Life Factor K_{HL} can be evaluated from table reported in fig. 4.32.

Duty cycles	Life factor
10,000 or fewer	1.5
Approx. 100,000	1.3
Approx. 10^6	1.15
10^7 or greater	1.0

Figure 4.32: Life Factor K_{HL} [1]

As for Bending Strength, the Life Factor is affected by the number of cycles along component's lifetime. When the number of cycles is unknown is assumed $K_{HL} = 1$.

- Lubricant Factor Z_L
  The Lubricant Factor Z_L is based upon the lubricant's kinematic viscosity at $50°\ C$. From graph in fig. 4.33 can be defined the value of Z_L.

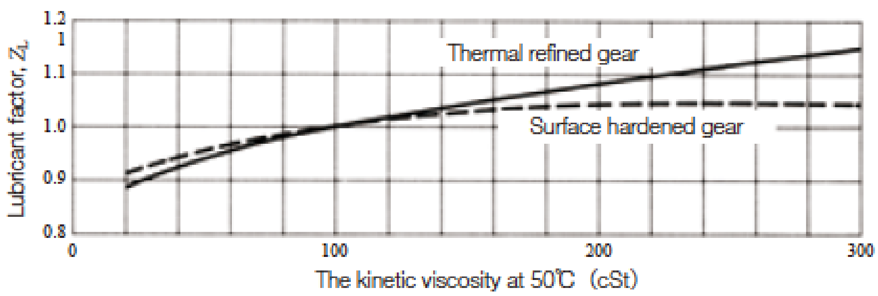


Figure 4.33: Lubricant Factor Z_L [1]

Kinematic viscosity is a measure of a fluid's internal resistance to flow under gravitational forces. It is determined by measuring the time in seconds, required for a fixed volume of fluid to flow a known distance by gravity through

a capillary within a calibrated viscometer at a closely controlled temperature. This value is converted to standard units such as centistokes (cSt) or square millimetres per second.[6]
Reporting viscosity without a reference to temperature is useless, the temperature must be defined to interpret the viscosity reading.[6]
Consulting different lubricant, available on the market, has been chosen an average value of Kinematic Viscosity, for this project of thesis, equal to 100 cSt. From graph in fig. 4.33, is defined a Lubricant Factor $Z_L = 1$.

- Surface Roughness Factor Z_R
  The Surface Roughness Factor Z_R is defined from the graph in figure 4.34. Z_R is function of the Average Roughness $R_{maxm} = \frac{R_{max1}+R_{max2}}{2}\sqrt[3]{\frac{100}{a}}$ $[\mu m]$. Parameters R_{max1} and R_{max2} are the surface roughness of the components coupled, a is the mounting distance.

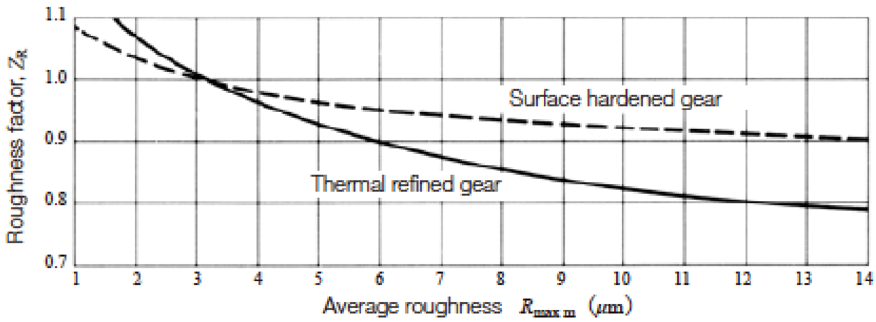


Figure 4.34: Surface Roughness Factor Z_R [1]

The Surface Roughness Factor is defined as reported in following table.

Table 4.15: Surface Roughness Factor Factor Z_R

Item	Pinion & Rack
Shift Input	1.043
Shift Output	1.082
Select Input	1.043
Select Output	1.06

- Lubrication Speed Factor Z_V
  The Lubrication Speed Factor Z_V is function of the tangential speed at the *reference diameter* v $[m/s]$. The value of Z_V is defined from graph in fig. 4.35.

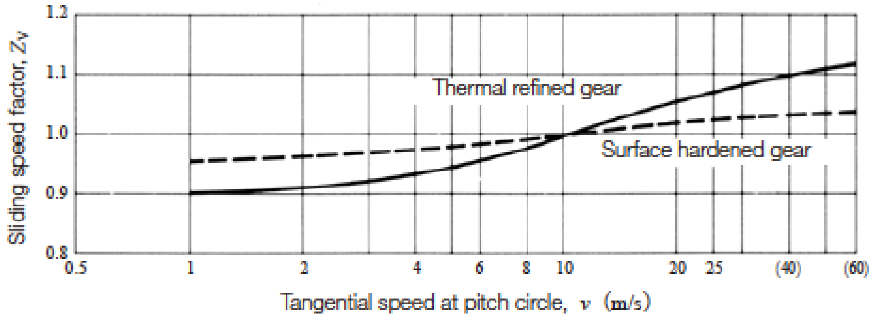


Figure 4.35: Lubrication Speed Factor Z_V [1]

In table 4.16 are reported the values adopted.

Table 4.16: Lubrication Speed Factor Z_V

Item	Pinion & Rack
Shift Input	0.8899
Shift Output	0.8911
Select Input	0.8897
Select Output	0.89

- Hardness Ratio Factor Z_W
  The Hardness Ratio Factor Z_W is applied only to quenched gears. The factor is calculated as follow

$$\begin{cases} 130 \leq H_B \leq 470 & Z_W = 1.2 - \frac{H_B - 130}{1700} \\ H_B \leq 130 \text{ } or \text{ } H_B \geq 470 & Z_W = 1 \end{cases} \tag{4.18}$$

- Size Factor K_{HX}
  The Size Factor K_{HX} typically is set $K_{HX} = 1$, as the condition affecting it are often unknown.

- Longitudinal Load Distribution Factor $K_{H\beta}$
  The Longitudinal Load Distribution Factor $K_{H\beta}$ is defined in two possible conditions:

  – Tooth Contact under load not predictable: this condition relates to the method of gear shaft support, and to the ratio facewidth/reference diameter b/d.

$\frac{b}{d_{01}}$	Method of gear shaft support			
	Bearings on both ends			Bearing on one end
	Gear equidistant from bearings	Gear close to one end (Rugged shaft)	Gear close to one end (Weak shaft)	
0.2	1.0	1.0	1.1	1.2
0.4	1.0	1.1	1.3	1.45
0.6	1.05	1.2	1.5	1.65
0.8	1.1	1.3	1.7	1.85
1.0	1.2	1.45	1.85	2.0
1.2	1.3	1.6	2.0	2.15
1.4	1.4	1.8	2.1	—
1.6	1.5	2.05	2.2	—
1.8	1.8	—	—	—
2.0	2.1	—	—	—

Figure 4.36: Longitudinal Load Distribution Factor $K_{H\beta}$ [1]

- Tooth Contact under load good: this condition furnishes a narrower range of values, $K_{H\beta} = 1 \sim 1.2$

In this project is considered a *good tooth contact*, and the parameter is set as $K_{H\beta} = 1$.

- Dynamic Load Factor K_V
  The Dynamic Load Factor K_V can be defined as in Strength Bending Validation, function of tooth profile precision and tangential speed at *reference diameter*, from table in figure 4.22.

- Overload Factor K_O
  The Overload Factor K_O can be defined as in Strength Bending Validation through equation 4.11 or table in figure 4.24. This parameter is set $K_O = 1$.

- Safety Factor for Pitting S_H
  The Safety Factor for Pitting S_H, is influenced by multiple environmental factors, and is not an easy parameter to be defined. Usually is advisable to use a factor at least $S_H \geq 1.5$.

- Allowable Hertz Stress $\sigma_{H,lim}$
  The Allowable Hertz Stress $\sigma_{H,lim}$, can be defined from tables depending on gear material used. The surface hardness, defined in tables, represents the value at the *pitch diameter*.

Material (Arrows indicate the ranges)		Surface hardness H_B	Surface hardness H_V	Lower limit of tensile strength kgf/mm2 (Reference)	σ_{Hlim} kgf/mm²
Cast steel	SC37			37	34
Cast steel	SC42			42	35
Cast steel	SC46			46	36
Cast steel	SC49			49	37
Cast steel	SCC3			55	39
Cast steel				60	40
Normalizing structural steel	S25C	120	126	39	41.5
Normalizing structural steel	S25C	130	136	42	42.5
Normalizing structural steel	S25C	140	147	45	44
Normalizing structural steel	S25C, S35C	150	157	48	45
Normalizing structural steel	S25C, S35C, S43C	160	167	51	46.5
Normalizing structural steel	S25C, S35C, S43C	170	178	55	47.5
Normalizing structural steel	S25C, S35C, S43C, S48C, S53C S58C	180	189	58	49
Normalizing structural steel	S35C, S43C, S48C, S53C S58C	190	200	61	50
Normalizing structural steel	S35C, S43C, S48C, S53C S58C	200	210	64	51.5
Normalizing structural steel	S35C, S43C, S48C, S53C S58C	210	221	68	52.5
Normalizing structural steel	S43C, S48C, S53C S58C	220	231	71	54
Normalizing structural steel	S43C, S48C, S53C S58C	230	242	74	55
Normalizing structural steel	S53C S58C	240	253	77	56.5
Normalizing structural steel	S53C S58C	250	263	81	57.5
Quenched and tempered structural steel	S35C	160	167	51	51
Quenched and tempered structural steel	S35C	170	178	55	52.5
Quenched and tempered structural steel	S35C	180	189	58	54
Quenched and tempered structural steel	S35C	190	200	61	55.5
Quenched and tempered structural steel	S35C, S43C	200	210	64	57
Quenched and tempered structural steel	S35C, S43C, S48C	210	221	68	58.5
Quenched and tempered structural steel	S35C, S43C, S48C	220	231	71	60
Quenched and tempered structural steel	S35C, S43C, S48C, S53C S58C	230	242	74	61
Quenched and tempered structural steel	S35C, S43C, S48C, S53C S58C	240	252	77	62.5
Quenched and tempered structural steel	S35C, S43C, S48C, S53C S58C	250	263	81	64
Quenched and tempered structural steel	S35C, S43C, S48C, S53C S58C	260	273	84	65.5
Quenched and tempered structural steel	S35C, S43C, S48C, S53C S58C	270	284	87	67
Quenched and tempered structural steel	S43C, S48C, S53C S58C	280	295	90	68.5
Quenched and tempered structural steel	S43C, S48C, S53C S58C	290	305	93	70
Quenched and tempered structural steel	S43C, S48C, S53C S58C	300	316	97	71
Quenched and tempered structural steel	S53C S58C	310	327	100	72.5
Quenched and tempered structural steel	S53C S58C	320	337	103	74
Quenched and tempered structural steel		330	347	106	75.5
Quenched and tempered structural steel		340	358	110	77
Quenched and tempered structural steel		350	369	113	78.5

Figure 4.37: $\sigma_{H,lim}$ Gear without Surface Hardening [1]

Material		Heat treatment before induction hardening	Surface hardness H_V (Quenched)	σ_{Hlim} kgf/mm²
Structural carbon steel	S43C S48C	Normalized	420	77
			440	80
			460	82
			480	85
			500	87
			520	90
			540	92
			560	93.5
			580	95
			600 and above	96
		Quenched and tempered	500	96
			520	99
			540	101
			560	103
			580	105
			600	106.5
			620	107.5
			640	108.5
			660	109
			680 and above	109.5
Structural alloy steel	SMn443 SCM435 SCM440 SNC836 SNCM439	Quenched and tempered	500	109
			520	112
			540	115
			560	117
			580	119
			600	121
			620	123
			640	124
			660	125
			680 and above	126

Figure 4.38: $\sigma_{H,lim}$ Induction Hardened Gear [1]

Material		Effective case depth (1)	Surface hardness H_V	σ_{Hlim} kgf/mm²
Structural carbon steel	S15C S15CK	Relatively shallow See NOTE (1)A	580	115
			600	117
			620	118
			640	119
			660	120
			680	120
			700	120
			720	119
			740	118
			760	117
			780	115
			800	113
Structural alloy steel	SCM415 SCM420 SNC420 SNC815 SNCM420	Relatively shallow See NOTE (1)A	580	131
			600	134
			620	137
			640	138
			660	138
			680	138
			700	138
			720	137
			740	136
			760	134
			780	132
			800	130
		Relatively thick See NOTE (1)B	580	156
			600	160
			620	164
			640	166
			660	166
			680	166
			700	164
			720	161
			740	158
			760	154
			780	150
			800	146

Figure 4.39: $\sigma_{H,lim}$ Carburized and Quenched Gear [1]

The evaluation of Allowable Hertz Stress, produces the following results, reported in table 4.17.

Table 4.17: Allowable Bending Stress at the Root $\sigma_{H,lim}$ $[MPa]$

Item	Pinion	Rack
Shift Input	612.91	505.04
Shift Output	612.91	505.04
Select Input	612.91	505.04
Select Output	612.91	505.04

After the selection of each parameter, related to rack-pinion system exploited, the calculation to validate the components, under Surface Durability targets are performed. In the following table are reported the results.

Table 4.18: Surface Durability Validation - Rack Pinion Shift Input

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	683.93	464.37
Allowable Torque $[Nm]$	13.68	—
Allowable Power $[KW]$	0.0716	0.0486

Table 4.19: Surface Durability Validation - Rack Pinion Shift Output

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	1476.68	1002.63
Allowable Torque $[Nm]$	59.06	—
Allowable Power $[KW]$	0.309	0.21

Table 4.20: Surface Durability Validation - Rack Pinion Select Input

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	683.57	464.13
Allowable Torque $[Nm]$	13.67	—
Allowable Power $[KW]$	0.057	0.04

Table 4.21: Surface Durability Validation - Rack Pinion Select Ouput

Item	Pinion	Rack
Allowable Circumferential Force $[N]$	953.54	647.43
Allowable Torque $[Nm]$	25.74	—
Allowable Power $[KW]$	0.108	0.073

The results of calculations shows that the rack-pinion chosen satisfy the requirements. The values obtained are all greater than the ones defined in section 2.3.3.

4.5.3 Dimensioning of Linear Guides for rack

The technical solution, chosen for linear motion of the rack, is the Recirculating Ball Guides. This solution presents the following advantages:

- Smooth low friction movement.
- High rigidity.
- Excellent load capacity.
- Easy re-grease system.
- Sealing system to prevent dirt ingress.
- Wide range of dimensions available.
- Wide range of materials available.
- Acceptable cost.

As reference for the choice of the Recirculating Ball Guides, is adopted HepcoMotion - Hepco Linear Ball Guides [7] catalogue.
The concept is to set the Sliding Block at zero dof, screwing it, to allow the rail sliding inside it. The rack, screwed on the rail able to slides along x direction on shift system, and y direction on select system.

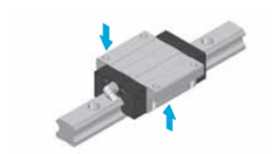
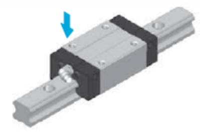
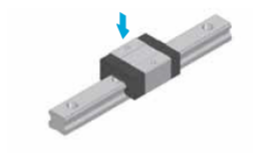
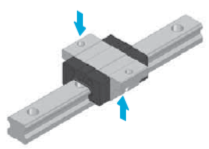
	**HLGF, HLGFL** Tapped hole in flange for mounting from above. Use bolt one size smaller when mounting from below.
	**HLGR, HLGRL** Tapped hole for mounting from above.
	**HLGSC, HLGSR** Tapped hole for mounting from above.
	**HLGSCF, HLGSF** Tapped hole in flange for mounting from above. Use bolt one size smaller when mounting from below.

Figure 4.40: HepcoMotion Linear Ball Guides[7]

In figure 4.40 are reported the typologies of recirculating ball guide available by HepcoMotion. The difference, is mainly related to the mounting alternatives of the sliding block. To exploit the system as explained previously, the solution code HLGF could be the best choice. Nevertheless is selected the solution code HLGR, more compact, the sliding block will be screwed on the plate.
First is selected the rail, with the same dimensions of the rack, then the corresponding sliding block is defined. The rack's width is 20 mm, the Rail selected is the one underlined in red in figure 4.42 from [7].

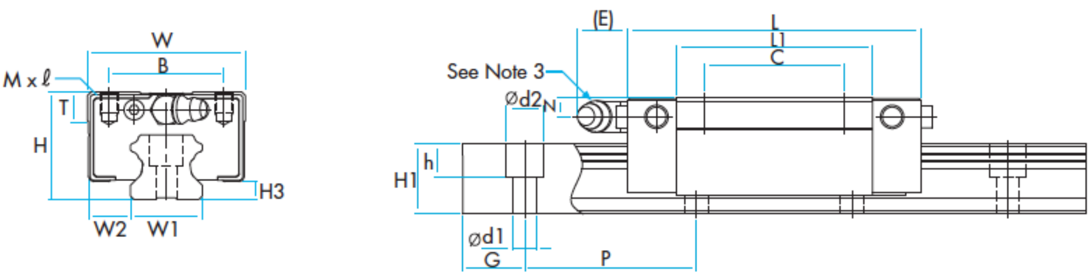


Figure 4.41: Linear Guide HLGR[7]

Ref No.	External Dimensions			Dimensions of HLG Block								H3
	Height H	Width W	Length L	B	C	M x	L1	T	N	E	Grease Nipple*3	
HLG15R	28	34	57	26	26	M4 x 5	40.8	6	10	6	A–M4	4.7
HLG15RL			65.3				49.1					
HLG20R	30	44	72.7	32	36	M5 x 6	53.1	8	7.5	12	B–M6F	6
HLG20RL			88.6		50		69					
HLG25R	40	48	83	35	35	M6 x 8	58.3	8	13	12	B–M6F	7
HLG25RL			102.9		50		78.2					
HLG30R	45	60	97.8	40	40	M8 x 10	70.8	8	10.3	12	B–M6F	7.5
HLG30RL			120		60		93					
HLG35R	55	70	110	50	50	M8 x 12	80.8	10	15	12	B–M6F	9
HLG35RL			135.4		72		106.2					
HLG45R	70	86	139	60	60	M10 x 17	101.9	15	20	16	B–PT1/8	10
HLG45RL			170.8		80		133.7					
HLG55R	80	100	163	75	75	M12 x 18	117.5	18	21	16	B–PT1/8	13
HLG55RL			201.1		95		155.6					

Ref No.	Dimensions of HLG Rail								Static Moment Capacity Nm			Weight	
	Width W1 ±0.05	W2	Height H1	Min G	Pitch P	d1 x d2 x h	C kN	Co kN	M	Mv	Ms	HLG Block kg	HLG Rail kg/m
HLG15R	15	9.5	13	10	60	4.5 x 7.5 x 5.3	9.9	16.2	115	115	129	0.18	1.3
HLG15RL							11.2	19.3	165	165	154	0.23	
HLG20R	20	12	16.5	10	60	6 x 9.5 x 8.5	14.9	23.9	221	221	251	0.31	2.2
HLG20RL							17.8	30.6	369	369	322	0.41	
HLG25R	23	12.5	20	10	60	7 x 11 x 9	22.1	33.1	337	337	398	0.53	3.0
HLG25RL							28.1	43.6	596	596	525	0.71	
HLG30R	28	16	26	12	80	9 x 14 x 12	33.0	57.1	711	711	828	0.9	4.85
HLG30RL							40.9	73.6	1203	1203	1067	1.1	
HLG35R	34	18	29	12	80	9 x 14 x 12	43.8	74.6	1062	1062	1298	1.5	6.58
HLG35RL							54.4	96.2	1797	1797	1674	2.01	
HLG45R	45	20.5	38	16	105	14 x 20 x 17	70.6	92.8	2257	2257	1796	2.89	11.03
HLG45RL							87.6	126.5	3781	3781	2448	3.74	
HLG55R	53	23.5	44	20	120	16 x 23 x 20	104.0	133.6	3810	3810	3094	4.28	15.26
HLG55RL							129.1	182.1	6410	6410	4220	5.59	

Figure 4.42: Linear Guide HLGR[7]

The main dimensions of Sliding Block and Rail, indicated in figure 4.42, are referred to the fig. 4.41.
In tables are reported also:

- Basic Dynamic Load Rating C $[N]$
- Basic Static Load Rating C_0 $[N]$
- Static Moment Capacity $M,\ M_v,\ M_s$ $[Nm]$

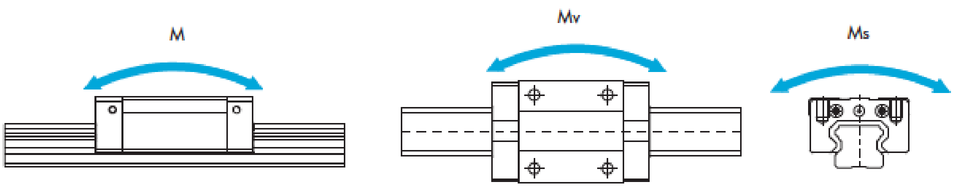


Figure 4.43: Linear Guide HLGR - Static Moment Capacity[7]

The value of major concerning is M_v of shift actuation, due to functioning principle of the actuation system. The linear guides, receives an axial force, on the rail, from force-stroke source, through the actuation cables (see introduction of section 4.4). As the select system is actuated, at the Knob Interface, is generated a reaction force. The reaction force, in y direction, generates a moment on the linear guide of shift's rack output. To evaluate forces acting on the linear guide, the following conditions are considered:

- Moment's Arm equal to distance from Knob interface to Rail end, $l_M = 0.36\ m$.
- Moment's Force perpendicular to rack axis (y direction)
- Moment's Force equal to Limit Tensile Pressure force (section 2.3.3), $F_M = 600\ N$.

The conditions considered are quite severe, to obtain a reliable system. The moment on the linear guide results to be $M_v = 216\ Nm$. The moment is lower than the one reported in manufacturer's tables, so the linear guide selected can be used in this application. Moreover the shift output rack, is the one that works under most severe conditions, its validation guarantees functionality of linear guides exploited on others racks.

Life Calculation

The Basic Dynamic Load Rating C of linear guides is evaluated as a constant directional load that provides 50 Km of linear travel. This is the distance at which 10% of guides will show signs of fatigue pitting of the block or rail tracks necessitating replacement. The 50 Km travel load rating shown in the catalogue is used to calculate the life of the system under normal operating conditions.[7]
Two factors are applied to the calculation:

- Contact Factor f_c: when two or more blocks are used on the same rail.
- Load Factor f_v: depending on type of load applied.

The life calculation can also be affected by:

- Excess load by inaccurate assembly.
- Contamination within the block.
- High speed short stroke motion with excessive load.
- Damage to the end plates.

$$L = \left(\frac{f_c C}{f_v P}\right)^3 50 \tag{4.19}$$

To calculate the Fatigue Life $[Km]$, the following conditions are imposed:

- Contact Factor f_c: evaluated from table in fig. 4.44, neglected for this calculation.
- Basic Dynamic Load Rating C: from figure 4.42 is defined $C = 14.9\ KN$.
- Load Factor f_v: necessary in case of additional forces from vibrations or impact. The highest travel speed is $v = 181.2\ mm/s = 10.87\ m/min$, so the Load Factor is defined as $f_v = 1.5$ for precaution.
- Applied Load P: considered $P = 130\ N$, as the fatigue life is mostly affected by a durability test (see section 2.3.3).

Contact Factor	Contact Factor (fc)
2	0.81
3	0.72
4	0.66
5	0.61
Over 6	0.60

Figure 4.44: Linear Guide - Contact Factor f_c [7]

Impact and Vibration Condition	Travel Speed Velocity (V)	fv
No External Impact or Vibration	V<= 15m / min Low Speed	1 - 1.5
Slight Impact and Vibration	15<V<= 60m / min Medium Speed	1.5 - 2.0
Medium Impact and Vibration	V>60m / min High Speed	2.0 - 3.5

Figure 4.45: Linear Guide - Load Factor f_v [7]

From calculation results a Life Fatigue $L > 22\ 10^6\ Km$, considering an actuation of $181.2\ mm$ (shift output), results a Life Fatigue $L = 123\ 10^9$ actuations.

Preload and Rigidity

Excessive static load can cause permanent deformation of the rolling element and raceway surface. The Static Load Rating C_0 is the mono-directional static load, at which occurs a permanent deformation of element and raceway surface, equivalent to 0.0001 times ball diameter.
Recirculating ball guides are designed in three pre-load levels. The pre-load, is used to eliminate clearance between the sliding block and rail surfaces, by the insertion of a ball larger than the space available. The rigidity of the block is function of the pre-load level and in normal applications Z_0 or Z_1 will suffice. For machining or higher impact applications Z_2 level is required.

Type	Preload type	Preload type	Equivalent preload force
HLG	Z0	Zero / Light (Stock Range)	0 – 0.03 x C
HLG	Z1	Light (Stock Range)	0.04 – 0.08 x C
HLG	Z2	Medium	0 .09 – 0.13 x C

Figure 4.46: Linear Guide - Preload Level [7]

Frictional Resistance

The frictional resistance is defined by the applied load and the Friction Factor. In lightly loaded applications, the seals can have a significant effect on the friction properties. The frictional resistance can be calculated as follow:

$$F = \mu P + f_s \tag{4.20}$$

The parameters assumed are:

- Frictional Factor μ: Considering a ratio $P/C = 0.008$ the value from the graph is $\mu \simeq 0.0075$.
- Seal Resistance: from table in fig. 4.48 is assumed $f_s = 4\ N$.
- Applied Load P: as for Life Fatigue $P = 130\ N$.

The calculation defines a value of Frictional Resistance $F = 4.97\ N$.

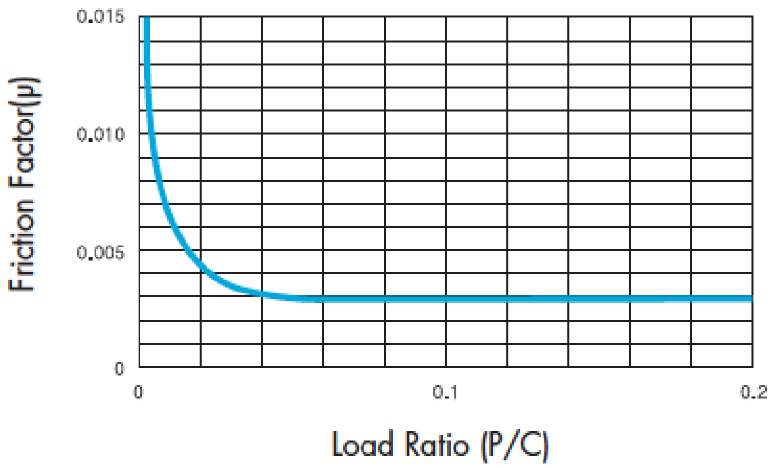


Figure 4.47: Linear Guide - Friction Factor f_s [7]

Ref No	Seal resistance (N) per block
HLG15	2N
HLG20	4N
HLG25	4N
HLG30	6N
HLG35	11N
HLG45	19N
HLG55	19N

Figure 4.48: Linear Guide - Seal Resistance [7]

4.5.4 Dimensioning of Rotative and Linear Bearings

In this section, is exposed the process for the selection of bearings. Two types are exploited in the project:

- Rotative Bearings: to guarantee rotation of pinions' shafts.
- Linear Bearing: to allow the shift system to change height along the arch.

The bearing selection process, is extrapolated from SKF - Rolling Bearing-17000EN [8].

The Rolling bearings are necessary, because they support and guide, with minimal friction, rotating or oscillating machine elements – such as shafts, axles or wheels – and transfer loads between machine components. Rolling bearings provide high precision and low friction and therefore enable high rotational speeds while reducing noise, heat, energy consumption, wear and are cost-effective.[8]
The two basic types of rolling element are: Ball Bearing and Roller Bearings. Balls and rollers differ on the kind of contact performed. The former define a *point contact*, which becomes an elliptical area when loaded, and provides low rolling friction (high speed with limited load supported). The latter define a *line contact*, which becomes likewise a rectangle when loaded, generating higher friction (low speed with high load supported).

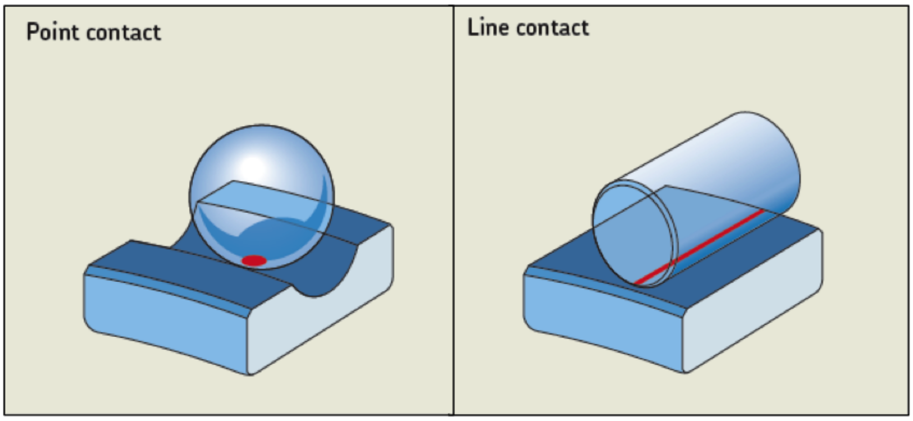


Figure 4.49: Rolling Bearing - Point vs. Line Contact[8]

Rolling bearings can be also classified into two groups based on the direction of the load they can support:

- Radial Bearings: can support mainly loads perpendicular to the shaft. Some solution is able to support purely radial loads, others can additionally support axial one. Contact angle of this solution is $\leq 45°$.

- Thrust Bearings: this solution presents contact angle $\geq 45°$. Thrust bearing can support mainly axial loads, but different technical solutions allow to react also to radial one. These bearings cannot work with same speed of radial one.

Now some terminology, in accordance with ISO 5593 Rolling bearings – Vocabulary, is reported.

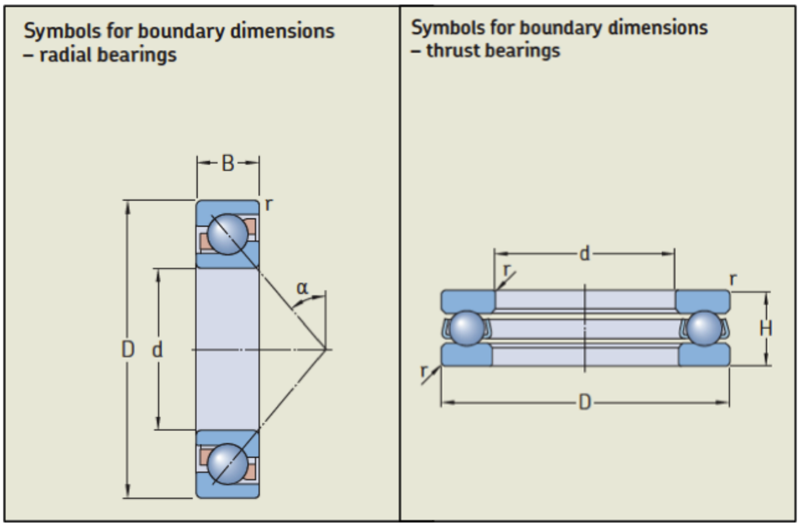


Figure 4.50: Rolling Bearing - Terminology[8]

- d Bore diameter
- D Outside diameter
- B Bearing width
- H Bearing height
- r Chamfer dimension
- α Contact angle

When selecting bearings, is important obtain the required level of performance, the lowest possible cost and robustness. The main factor to be considered are:

- Lubricant and Supply method
- Shaft and Housing
- Bearing clearance class
- Cage material and Guidance
- Dimensional stability
- Precision requirements
- Bearing sealing

- Mounting method and Maintenance

The selection process recommended by SKF, follows the steps in fig. 4.51.

Figure 4.51: Rolling Bearing - Selection Process[8]

Actually the selection process doesn't move straight step by step, however exists an interdependence between them.

Performance and Operating conditions

The first step of bearing selection, requires the evaluation of the following factors, as much detailed as possible: bearing life, speed capability, ability to withstand applied acceleration level, precision of radial and axial position of the shaft, ability to cope with low or high temperatures, generated noise and vibration levels. The relative importance of these performance factors, can influence the nature of the path taken through the steps of the bearing selection and application analysis process.[8] The most important operating parameters are:

- Load
- Speed
- Temperature
- Lubricant

Usually these can be determined from physical and mechanical analysis of the application, or from experience with similar applications. Operating conditions typically vary over time, e.g. in variable speed applications or because of seasonal temperature changes or increased output power.[8]
The relationship between principal operating conditions, application requirements and various aspects of a bearing arrangement's design are shown in figure 4.52, to obtain a robust and cost- effective solution.

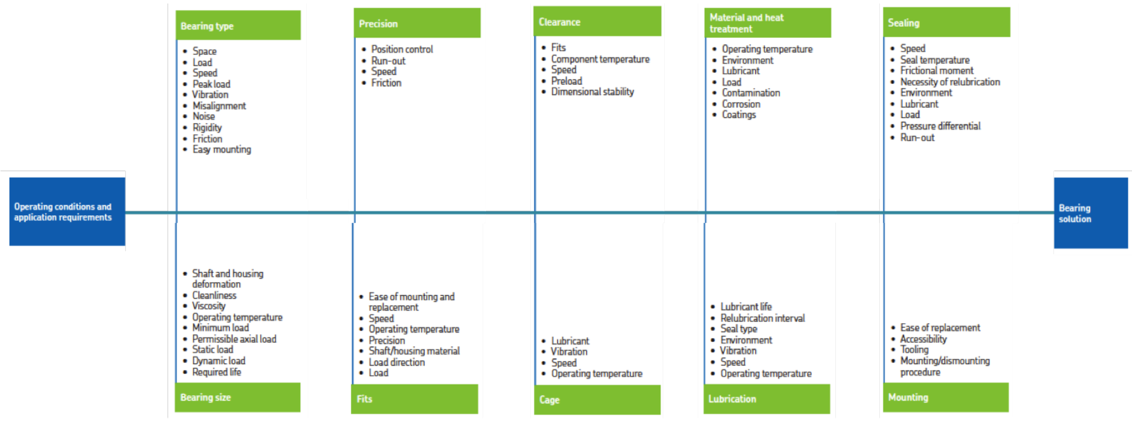


Figure 4.52: Rolling Bearing - Operating Conditions[8]

Bearing Type and Arrangement

In this step is defined the relationship between bearing properties and different applications, is important to consider, when selecting a bearing, the arrangement and the types of bearing to use with it.
A bearing arrangement supports and locates a shaft, radially and axially, relative to other components such as housings. Typically, two bearing supports are required to position a shaft. Depending on certain requirements, such as stiffness or load directions, a bearing support may consist of one or more bearings. Bearing arrangements comprising two bearing supports are:[8]

- Locating/Non-locating bearing arrangements
- Adjusted bearing arrangements
- Floating bearing arrangements

In following table, from SKF - Rolling Bearing-17000 EN, are reported the technical characteristics of each type of bearing solution.

Suitability of rolling bearings for industrial applications

Symbols

- +++ excellent
- ++ good
- \+ fair
- \- poor
- -- unsuitable
- ⇔ double direction
- ← single direction
- □ non-locating displacement on the seat
- ■ non-locating displacement within the bearing
- ✓ yes
- ✗ no

Bearing type	Variants	Load carrying capability: Radial load	Load carrying capability: Axial load	Load carrying capability: Moment load	Misalignment: Static misalignment	Misalignment: Dynamic misalignment (few tenths of a degree)
Deep groove ball bearings	A, B	+	+ ⇔	A –, B +	–	– –
Insert bearings	A, B, C	+	+ ⇔	– –	++	– –
Angular contact ball bearings, single row		+1)	++ ←	– –	–	– –
matched single row	A, B, C	A, B ++ C ++1)	A, B ++ ⇔ C ++ ←	A ++, B + C – –	A, C – –, B –	– –
double row	A, B	++	++ ⇔	++	– –	– –
four-point contact		+1)	++ ⇔	– –	– –	– –
Self-aligning ball bearings		+	–	– –	+++	+2)
Cylindrical roller bearings, with cage	A, B	++	– –	– –	–	– –
	A, B, C, D	++	A, B + ← C, D + ⇔	– –	–	– –
full complement, single row	A, B	+++	+ ←	– –	–	– –
full complement, double row	A, B, C, D	+++	A – –, B + ← C + ⇔	– –	–	– –
Needle roller bearings, with steel rings	A, B, C	++	– –	– –	A, B – C ++	– –
assemblies / drawn cups	A, B, C	++	A, B – – C –	– –	–	– –
combined bearings	A, B, C	++	A –, B + C ++	– –	– –	– –
Tapered roller bearings, single row		+++1)	++ ←	– –	–	– –
matched single row	A, B, C	A, B +++ C +++1)	A, B ++ ⇔ C ++ ←	A +, B ++ C – –	A – B, C – –	– –
double row	A, B	+++	++ ⇔	A + B ++	A –, B – –	– –
Spherical roller bearings		+++	+ ⇔	– –	+++	+2)
CARB toroidal roller bearings, with cage		+++	– –	–	++	–
full complement		+++	– –	–	++	–
Thrust ball bearings	A, B	– –	A + ← B + ⇔	– –	– –	– –
with sphered housing washer	A, B	– –	A + ← B + ⇔	– –	++	– –
Cylindrical roller thrust bearings		– –	++ ←	– –	– –	– –
Needle roller thrust beairngs		– –	++ ←	– –	– –	– –
Spherical roller thrust bearings		+1)	+++ ←	– –	+++	+2)

1) Provided the F_a/F_r ratio requirement is met 2) Reduced misalignment angle – contact SKF 3) Depending on cage and axial load level

Figure 4.53: Rolling Bearing - Industrial Application[8]

Arrangement				Suitable for					Design features			
Locating	Non-locating	Adjusted	Floating	Long grease life	High speed	Low run-out	High stiffness	Low friction	Integral sealing	Separable ring mounting	Tapered bore	Standard housings and accessories available
⊖	□	✗	✓	A +++ B ++	A +++ B +	A +++ B ++	+	+++	A ✓	✗	✗	✗
⊖	⊖	✗	✗	+++	++	A, B + C ++	+	++	✓	✗	✗	✓
✗	✗	✓	✗	++	++	+++	++	++	✓	✗	✗	✗
A, B ⊖ C ←	A, B □ C ✗	✗	✗	++	++	+++	++	++	✗	✗	✗	✗
↔	□	✗	✗	++	++	++	++	++	A ✓	B ✓	✗	✗
⊖ 1)	– –	– –	– –	+	+++	++	++	++	✗	✓	✗	✗
⊖	□	✗	✓	+++	++	++	+	+++	✓	✗	✓	✓
✗	■	✗	✗	++	+++	+++	++	+++	✗	✓	✗	✗
A, B ← C, D ⊖	A, B ■ ← C, D ✗	✗	A ✓ B, C, D ✗	++ 3)	+++	++	++	+++	✗	✓	✗	✗
←	A, B ←	✗	✓	–	+	+	+++	–	✗	A ✗ B ✓	✗	✗
B ← C, D ⊖	A ■ ⊖ B ■ ←	✗	✗	–	+	+	+++	–	D ✓	✗	✗	✗
✗	■ ⇔	✗	✗	++	++	+	++	+	A ✓	✓	✗	✗
A, B ✗ C ←	A, B ■ C ■ ←	✗	✗	++	++	+	++	+	B, C ✓	✓	✗	✗
←	✗	✓	✗	+	+	+	++	+	✗	✓	✗	✗
←	✗	✓	✗	+	++	+++	++	+	✗	✓	✗	✗
A, B ⊖ C ←	A, B □ C ✗	A, B ✗ C ✓	✗	+	+	++	+++	+	✗	✓	✗	✗
⇔	□	✗	✗	+	+	++	+++	+	✓	✓	B ✓	✗
⇔	□	✗	✓	+	++	+++	++	+	✓	✗	✓	✓
✗	■	✗	✗	+	++	+++	++	+	✗	✗	✓	✓
✗	■	✗	✗	–	+	+++	++	–	✓	✗	✓	✓
A ← B ⊖	✗	✗	✗	+	–	++	+	+	✗	✓	✗	✗
A ← B ⊖	✗	✗	✗	+	–	+	+	+	✗	✓	✗	✗
←	✗	✗	✗	–	–	+	+++	+	✗	✓	✗	✗
←	✗	✗	✗	–	–	+	+++	+	✗	✓	✗	✗
←	✗	✓	✗	–	+	+	+++	+	✗	✓	✗	✗

Figure 4.54: Rolling Bearing - Industrial Application[8]

For the choice of bearing type and arrangement, the selection criteria reported by SKF are the following:

- Available Space: often the boundary dimensions of a bearing are predetermined by the machine's design. Typically, the shaft diameter determines the bearing bore diameter. For the same bore diameter, different outside diameters and widths may be available.[8]

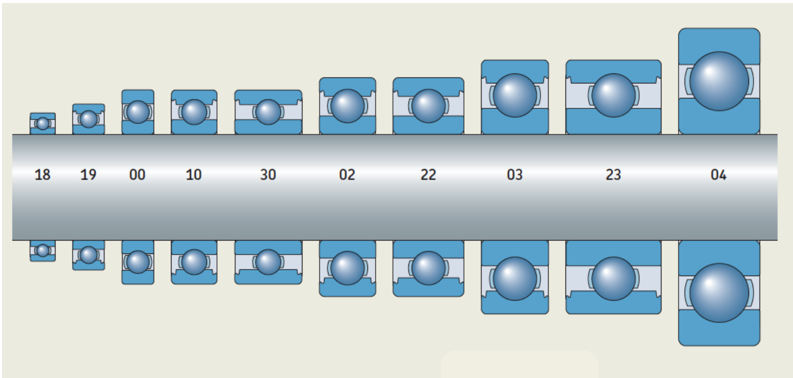


Figure 4.55: Rolling Bearing - ISO dimension series for same bore diameter[8]

- Loads: the direction and module of loads, are of primary importance in bearings type selection. Where the load on a bearing is a combination of radial and axial load, the summation of the components determines the direction of the combined load. The suitability of a bearing for a certain direction of load corresponds to its contact angle, the greater the contact angle, the higher the axial load carrying capacity of the bearing.[8]
  In figure 4.57, is reported the recommended selection of bearings, function of the load direction.

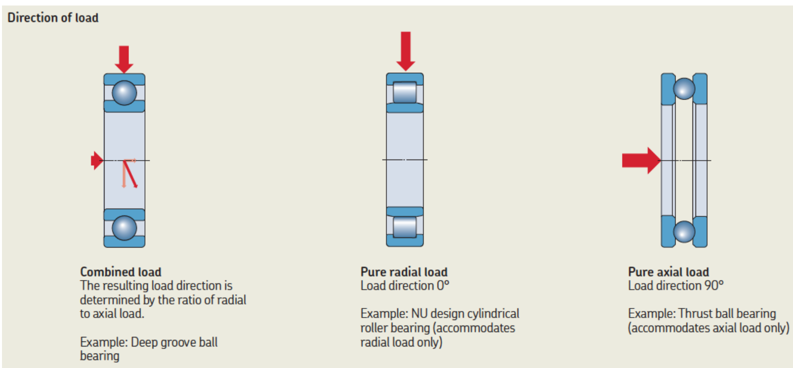


Figure 4.56: Rolling Bearing - Load Direction[8]

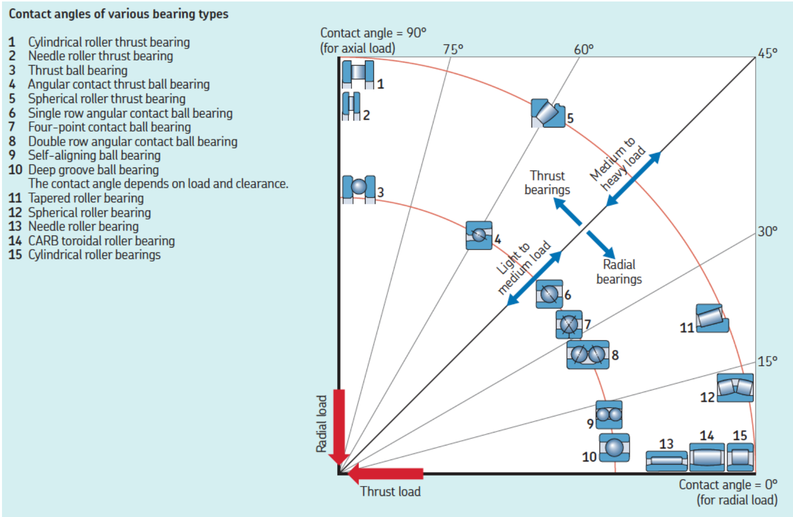


Figure 4.57: Rolling Bearing - Contact Angle[8]

- Speed and Friction: the allowed operating temperature of rolling bearings, defines limits on the speed achievable. The operating temperature is determined by the frictional heat generated in the bearing, except in machines where process heat is dominant. The following considerations are useful in bearings selection process: Ball bearings have lower frictional moment than Roller bearings, Radial bearings support higher speed than Thrust bearings, Single row bearings generate lower frictional heat than Double/Multi-row bearings, Ceramic bearings support higher speed than steel.

- Misalignment: bearing types vary in their ability to compensate for misalignment between the shaft and housing. The types of misalignment are reported in figure 4.58, the types of bearings more apt to balance misalignment are reported in fig. 4.53 and 4.54.

- Temperature: the allowable operating temperature of Rolling bearings is influenced by: the dimensional stability or rings and rolling elements, the cage, the seals and the lubricants.

- Stiffness: the stiffness of a rolling bearing, is characterized by the magnitude of the elastic deformation, in the bearing, under load. It depends not only on bearing type, but also on bearing size and operating clearance.[8]
  When selecting a bearing is useful remember: Roller bearings are stiffer than Ball bearings, Hybrid bearings are stiffer than all-steel bearings, pre-loads increase bearings' stiffness.

- Mounting and Dismounting: the selection, dependent on this parameter, is

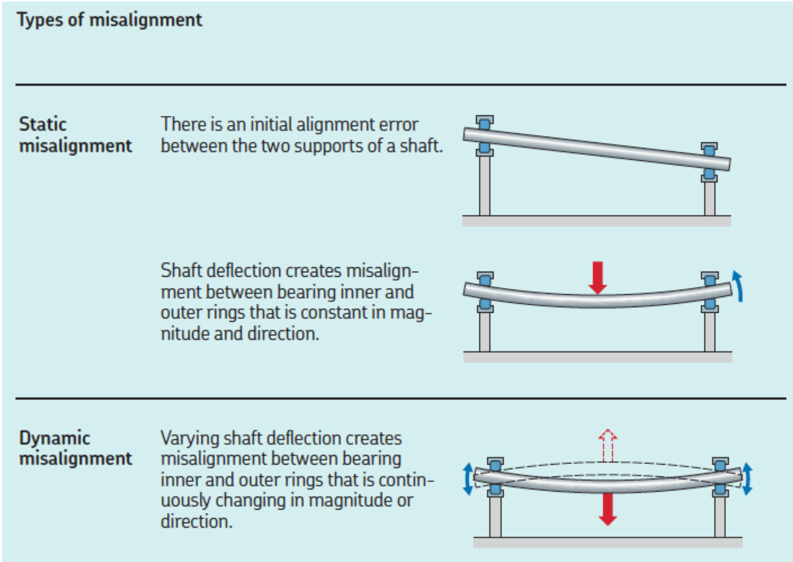


Figure 4.58: Rolling Bearing - Misalignment[8]

a matter of convenience during mounting or dismounting process. When is preferable to mount inner and outer ring independently a Separable bearing is advised. A bearing with tapered bore is advisable with tapered seat or sleeve. Again specifications for each type of bearing is available in fig. 4.53 and 4.54.

- Cost: once defined the bearing type and arrangement, the final choice is affected also by the cost. Bearings with high level of availability generally provide a cost-effective solution.

Bearing Size

The size of a bearing, must be sufficient to ensure that it is strong enough to deliver the required/expected life under defined operating conditions. A bearing can be viewed as a system of components: raceways, rolling elements, cage, seals (if present) and lubricant. The performance of each component contributes to or determines the performance and life of the bearing.[8]
The operating conditions define the main parameter to consider between: RFC (rolling contact fatigue), Permanent deformation (rolling element or raceway), Cage type/material, Speed limit, Lubricant properties. To define the correct bearing size, there are two possible methods:

- Size selection based on Rating Life: based on the required bearing life, taking into account the possible effects of rolling contact fatigue. Valid for typical operating conditions (normal speed, good lubrication, low loads).

The estimation of expected bearing life, can be performed through the basic rating life L_{10} or the SKF rating life L_{10m} (more severe). Bearing life is defined by SKF as the number of revolutions (or the number of operating hours) at a given speed that the bearing is capable of enduring, before the first sign of metal fatigue (spalling) occurs on a rolling element, or on the raceway of the inner/outer ring. The Basic rating life of a bearing, in accordance with ISO 281 is

$$L_{10} = \left(\frac{C}{P}\right)^p \tag{4.21}$$

When the speed is constant, is preferable to calculate the Basic rating life in hours

$$L_{10h} = \frac{10^6}{60n} L_{10} \tag{4.22}$$

$$\text{With} \begin{cases} C = \text{Basic dynamic load rating } [KN] \\ P = \text{Equivalent dynamic bearing load } [KN] \\ n = \text{Rotational speed } [rpm] \\ p = \text{Exponent of life equation} \begin{cases} p = 3 \text{ for ball bearings} \\ p = 10/3 \text{ for roller bearings} \end{cases} \end{cases}$$

For modern high-quality bearings, the calculated basic rating life can deviate significantly from the actual service life in a given application. Service life in a particular application depends not only on load and bearing size, but also on a variety of influencing factors including lubrication, degree of contamination, proper mounting and other environmental conditions.[8]
To supplement the basic rating life, a Life modification factor a_{SKF} is exploited, which allows to calculate the SKF rating life

$$L_{nm} = a_1 a_{SKF} L_{10} \tag{4.23}$$

With a_1 = life adjustment factor for reliability, defined from fig. 4.59, which becomes less relevant for load levels lower than the fatigue load limit $P < P_u$.

Values for life adjustment factor a_1

Reliability	Failure probability n	SKF rating life L_{nm}	Factor a_1
%	%	million revolutions	–
90	10	L_{10m}	1
95	5	L_{5m}	0,64
96	4	L_{4m}	0,55
97	3	L_{3m}	0,47
98	2	L_{2m}	0,37
99	1	L_{1m}	0,25

Figure 4.59: Rolling Bearing - Life Adjustment Factor a_1 [8]

The basic dynamic load rating C is used for calculating basic rating life and SKF rating life, for bearings that rotate under load. The C value is defined as: the bearing load that will result in an ISO 281 basic rating life of 1 000 000 revolutions. It is assumed that the load is constant in magnitude and direction and is radial for radial bearings and axial, centrally acting, for thrust bearings. The basic dynamic load ratings for SKF bearings are determined in accordance with the procedures outlined in ISO 281, and apply to bearings made of chromium bearing steel, heat treated to a minimum hardness of 58 HRC, operating under normal conditions.[8]
When calculating the bearing rating life, a value for equivalent dynamic bearing load P is required for both basic bearing life and SKF bearing life equations. The loads acting on a bearing are calculated according to the laws of mechanics using the external forces – such as forces from power transmission, work forces, gravitational or inertial forces – that are known or can be calculated. In real-world circumstances, the loads acting on a bearing may not be constant, can act both radially and axially, and are subject to other factors that require the load calculations to be modified or, in some cases, simplified.[8]
P is defined by SKF as the hypothetical load, constant in magnitude and direction, that acts radially on radial bearings and axially and centrally on thrust bearings. This hypothetical load, when applied, would have the same influence on bearing life as the actual loads to which the bearing is subjected (fig. 4.60).
When a bearing is loaded with both radial load F_r and axial load F_a, are constant in magnitude and direction, the equivalent dynamic bearing load P can be obtained from equation

$$P = XF_r + YF_a \tag{4.24}$$

With $\begin{cases} F_r = \text{Radial bearing load } [KN] \\ F_a = \text{Axial bearing load } [KN] \\ X = \text{Radial load factor} \\ Y = \text{Axial load factor} \end{cases}$

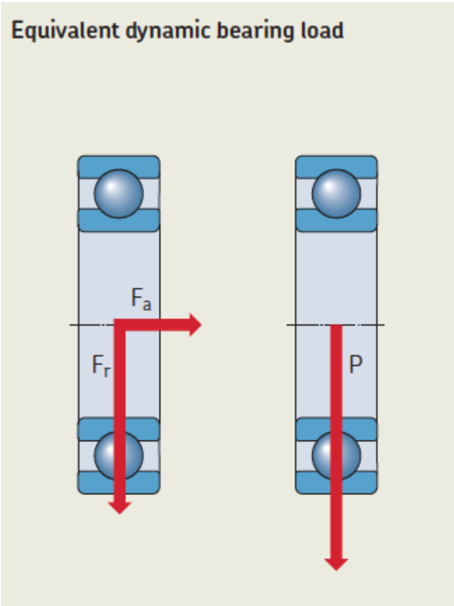


Figure 4.60: Rolling Bearing - Equivalent Dynamic Bearing Load P [8]

Axial load only influences the equivalent dynamic load P in single row radial bearing, if the ratio F_a/F_r exceeds a certain limiting factor e. With double row bearings, even light axial loads influence the equivalent load and have to be considered. The same general equation also applies to spherical roller thrust bearings, which can support both axial and radial loads.
With geared transmissions, the theoretical tooth forces can be calculated from the power transmitted and the design characteristics of the gear teeth. However, there are additional dynamic forces, produced either by the gear, or by the input or output shaft. Additional dynamic forces from gears can be the result of pitch or form errors of the teeth and from unbalanced rotating components. Gears produced to a high level of accuracy have negligible additional forces. For lower precision gears, use the following gear load factors:

- pitch and form errors $< 0.02\ mm$: $[1.05 \div 1.1]$
- pitch and form errors $[0.02 \div 0.1]\ mm$: $[1.1 \div 1.3]$

Additional forces arising from the type and mode of operation of the machines that are coupled to the transmission can only be determined when the operating conditions, the inertia of the drive line and the behaviour of couplings or other connectors are known. Their influence on the rating lives of the bearings is included by using an operation factor that takes into account the dynamic effects of the system.[8]
The Life modification factor a_{SKF}, takes into account three important factors:

- Fatigue load limit, in relation to the acting bearing equivalent load (P_u/P).
- Effect of the contamination level in the bearing η_c.
- lubrication condition (viscosity ratio κ).

To estimate a_{SKF}, as function of operating conditions, is useful the graph reported in figure 4.61. The abscissa axis represents the combined influence of load and contamination on fatigue. The viscosity ratio, κ, represents the lubrication conditions and their influence on fatigue.

- Area A: characterized by very high load and/or severe indentations. The lubricating conditions in this area can only marginally improve the expected fatigue life, improvements depend on the relationship between the contamination level and the load level P_u/P. To improve SKF rating life, either the load must be reduced, or the cleanliness must be improved.
- Area B: characterized by higher life modification factors, which influences positively the Basic rating life, increasing SKF rating life. On this domain small deviations from estimated load level, cleanliness factor and lubrication conditions will greatly affect the life modification factor.
- Area C: characterized by lower sensitivity of a_{SKF} to variations. Deviations from estimated load level, cleanliness factor and lubrication conditions will not substantially affect the value of Life modification factor, more robust SKF rating life. The load level ranges in area C are:
  * $P_u \leq P \leq 0.5C$ Ball bearing
  * $P_u \leq P \leq 0.33C$ Roller bearing

  Some useful issues:

  * Improve cleanliness (better sealing, infiltration and assembly conditions) increases the contamination factor η_c.
  * Cooling or using a lubricant with higher viscosity increases the viscosity ratio κ.
  * Choose a larger bearing size increases the ratio P_u/P (and the basic rating life L_{10}).

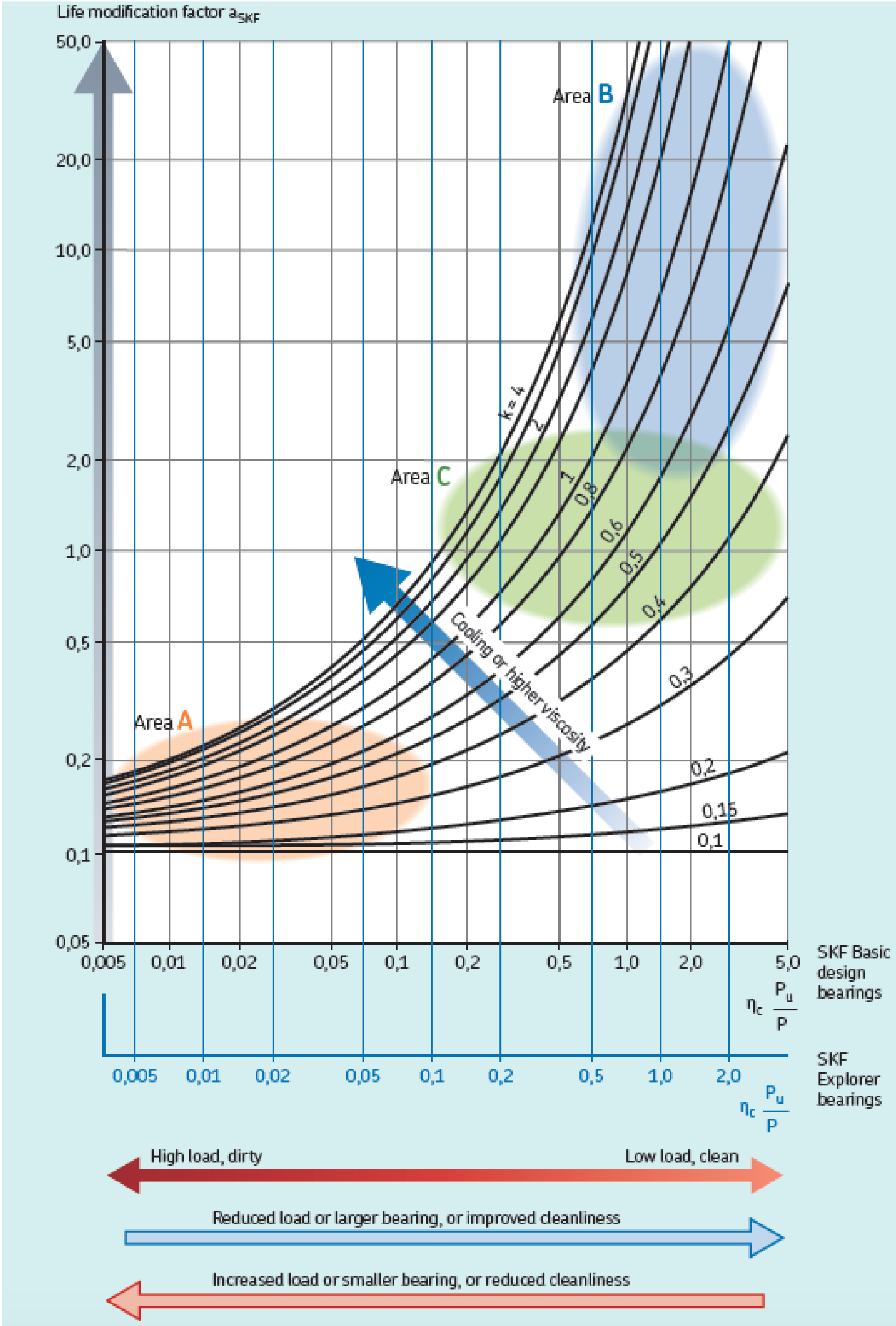


Figure 4.61: Rolling Bearing - Life Modification Factor a_{SKF} [8]

When a bearing has reached its normal speed and operating temperature, the lubrication condition of the bearing is

$$\kappa = \frac{\nu}{\nu_1} \tag{4.25}$$

With $\begin{cases} \kappa = \text{Viscosity Ratio} \\ \nu = \text{Actual operating viscosity of the lubricant } [mm^2/s] \\ \nu_1 = \text{Rated viscosity } [mm^2/s] \end{cases}$

The Actual operating viscosity can be define from graph in fig. 4.62, while ν_1 through the graph in fig. 4.63, function of mean bearing diameter and rotational speed - mean diameter of the bearing $d_m = 0.5(d + D)$ $[mm]$.

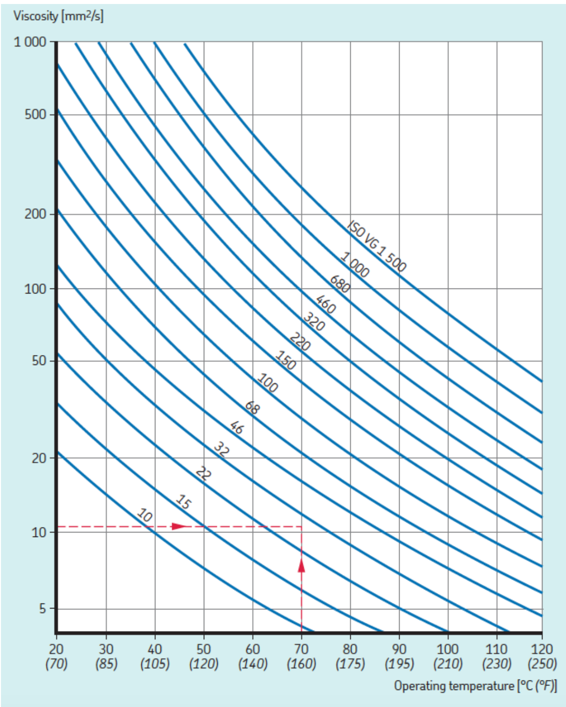


Figure 4.62: Rolling Bearing - Actual Operating Viscosity ν [8]

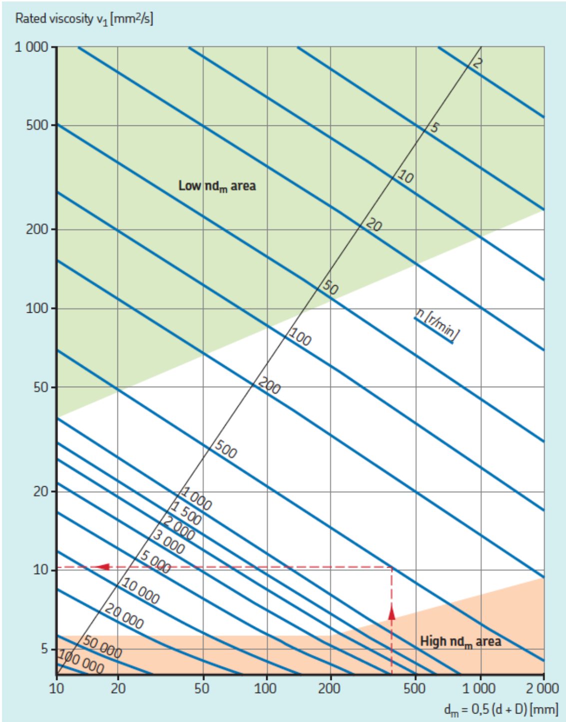


Figure 4.63: Rolling Bearing - Rated Viscosity ν_1 [8]

The higher the κ value, the better the lubrication condition of the bearing and its expected rated life. Most of the bearings are designed for a lubrication condition ranging from $[1 \div 4]$

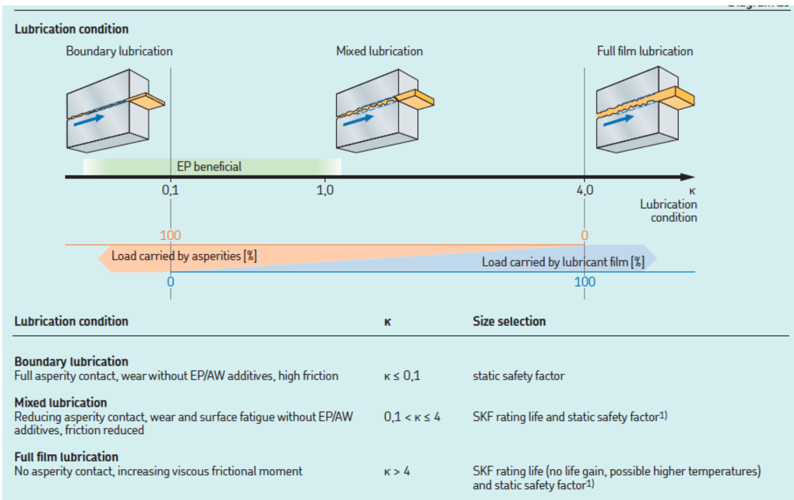


Figure 4.64: Rolling Bearing - Viscosity Ratio κ [8]

For values $\kappa < 1$, SKF recommends a Size selection based on Static Load (see next point).
The contamination factor η_c, takes into account how the level of solid particle contamination, of the lubricant, influences the calculated bearing fatigue life. The particles cause indentations in the rolling surfaces of the bearing, and these indentations increase the local contact stress, which reduces the expected fatigue life (fig.4.65).

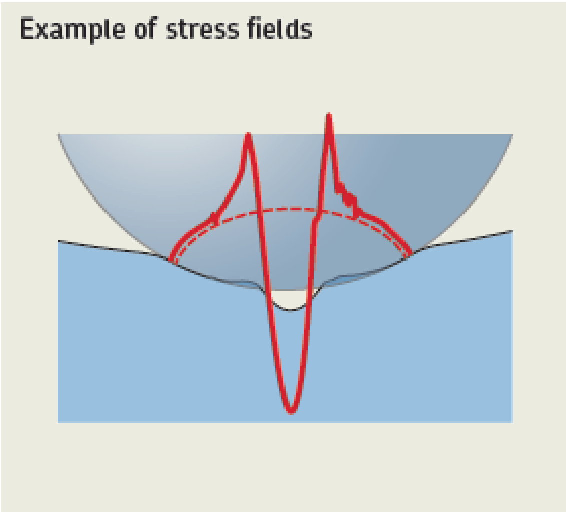


Figure 4.65: Rolling Bearing - Stress Field [8]

$$\begin{cases} \eta_c = 1 \text{ Perfectly clean conditions} \\ \eta_c \to 0 \text{ Severely contaminated conditions} \end{cases}$$

In SKF rating life model, the contamination factor for a certain bearing acts as a stress raiser, by reducing the bearing fatigue load limit P_u (load level below which metal fatigue will not occur). The stress-raising influence of contamination on bearing fatigue depends on a number of parameters, including: bearing size, relative lubricant condition, size and distribution of solid contaminant particles and types of contaminants.[8] Guideline values in accordance with ISO 281 are listed in the following.

Guideline values for factor η_c for different level of contamination

Conditions	Factor η_c[1)] for bearings with diameter $d_m < 100$	$d_m \geq 100$ mm
Extreme cleanliness • Particle size of the order of the lubricant film thickness • Laboratory conditions	1	1
High cleanliness • Oil filtered through an extremely fine filter • Typical conditions: sealed bearings that are greased for life	0,8 ... 0,6	0,9 ... 0,8
Normal cleanliness • Oil filtered through a fine filter • Typical conditions: shielded bearings that are greased for life	0,6 ... 0,5	0,8 ... 0,6
Slight contamination • Typical conditions: bearings without integral seals, coarse filtering, wear particles and slight ingress of contaminants	0,5 ... 0,3	0,6 ... 0,4
Typical contamination • Typical conditions: bearings without integral seals, coarse filtering, wear particles, and ingress from surroundings	0,3 ... 0,1	0,4 ... 0,2
Severe contamination • Typical conditions: high levels of contamination due to excessive wear and/or ineffective seals • Bearing arrangement with ineffective or damaged seals	0,1 ... 0	0,1 ... 0
Very severe contamination • Typical conditions: contamination levels so severe that values of ηc are outside the scale, which significantly reduces the bearing life	0	0

Figure 4.66: Rolling Bearings - Contamination Factor η_c [8]

- Size selection based on Static Load: based on the static load that the bearing can support, taking into account the possible effects of permanent deformation, and requires calculation of the static safety factor. Valid for applications with the following conditions:

  - Bearing not rotating and subjected to continuous high load or intermittent peak loads.
  - Bearing makes slow oscillating movements under load.
  - Bearing rotates and, in addition to the normal fatigue, has to sustain temporary high peak loads.
  - Bearing rotates under load at low speed ($n < 10\ rpm$) and is required to have only a limited life.

  In general very low speed (or static), bad lubrication conditions, occasional peak loads require a selection based on static load.
  Loads comprising radial and axial components, are to be evaluated in relation to the static load rating C_0, and must be converted into an equivalent static bearing load. This is defined as the hypothetical load which, when applied, would cause the same maximum rolling element load in the bearing as the actual loads to which the bearing is subjected.[8] The Equivalent static bearing load is defined with the following equation[8]

$$P_0 = X_0 F_r + Y_0 F_a \tag{4.26}$$

$$\text{With} \begin{cases} F_r = \text{Radial bearing load } [KN] \\ F_a = \text{Axial bearing load } [KN] \\ X_0 = \text{Radial load factor} \\ Y_0 = \text{Axial load factor} \end{cases}$$

Once calculated P_0 is possible define the Static Safety Factor s_0.

$$s_0 = C_0/P_0 \tag{4.27}$$

As alterative s_0 can be evaluated from tables in figure 4.67, where are reported guideline values based on experience.

Guideline values for the static safety factor s_0 – for continuous and/or occasional loads – ball bearings

Certainty of load level	**Continuous motion** Permanent deformation acceptance Yes	Some	No	**Infrequent motion** Permanent deformation acceptance Yes
High certainty For example, gravity loading and no vibration	0,5	1	2	0,4
Low certainty For example, peak loading	≥ 1,5	≥ 1,5	≥ 2	≥ 1

Guideline values for the static safety factor s_0 – for continuous and/or occasional loads – roller bearings[1)]

Certainty of load level	**Continuous motion** Permanent deformation acceptance Yes	Some	No	**Infrequent motion** Permanent deformation acceptance Yes
High certainty For example, gravity loading and no vibration	1	1,5	3	0,8
Low certainty For example, peak loading	≥ 2,5	≥ 3	≥ 4	≥ 2

Figure 4.67: Rolling Bearing - Static Safety Factor s_0 [8]

In applications where the load is not the main parameter affecting bearing size selection, the bearing may be lightly loaded in relation to its size and carrying capacity. With very light loads, skidding and smearing of raceways or cage damage, acquire grater importance. Under this conditions, rolling bearings must always be subjected to a given minimum load dimensioning. As general rule, minimum loads of $0.01C$ should be imposed on ball bearings, and $0.02\ C$ on roller bearings. The importance of minimum load is greater in applications where there are rapid accelerations or rapid starts and stops, and where speeds exceed 50% of the limit speeds.

Lubrication

Rolling bearings must be adequately lubricated to be reliable. The lubricant is required to reduce friction, inhibit wear, protect the bearing surfaces against corrosion and may also be needed to provide cooling.[8] The first step in the lubrication selection process, is to decide whether to use grease or oil. The flow chart, shown in fig. 4.68, help to select the correct lubrication method.

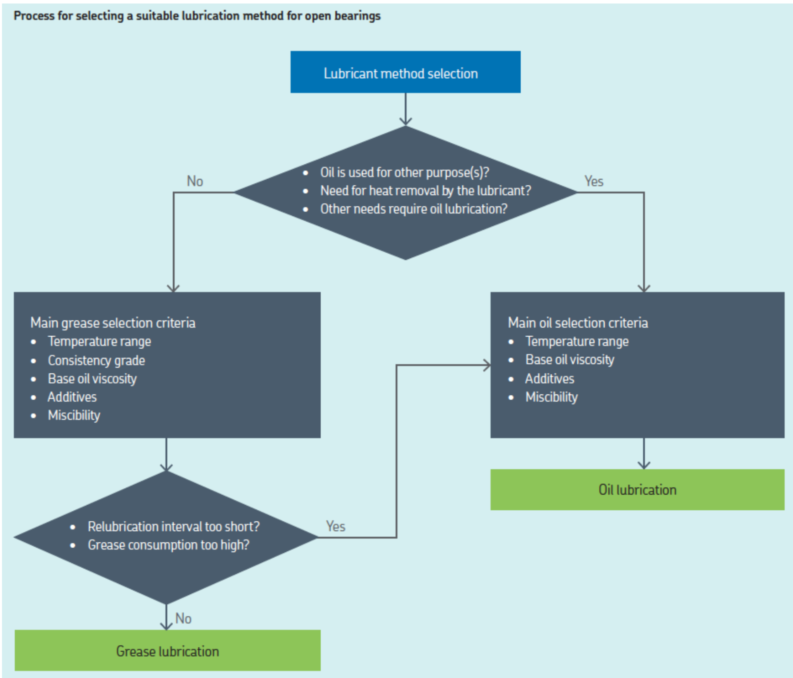


Figure 4.68: Rolling Bearing - Lubrication Method Selection [8]

Tipically grease is adopted as lubrication method, due to its cost-effectiveness, the exceptions are for applications with: very short relubrication interval, lubricating oil already implemented for other system, heat removal requirements, cost increase related to grease use.
Grease life and Relubrication interval depend mainly on: bearing type and size, speed, load ratio C/P, operating temperature, grease type. Relubrication should occur frequently enough to avoid grease deterioration affecting bearing life. SKF relubrication interval, t_f, can be defined from graph in figure 4.69, as function of: rotational speed n $[rpm]$, bearing mean diameter d_m $[mm]$, bearing factor b_f, load ratio C/P. In this project of thesis the temperature, results to be the parameter of main importance in lubricant selection. The thermal cycles to which the test bench is subjected, influence the choice of the lubricant, to obtain a reliable solution.

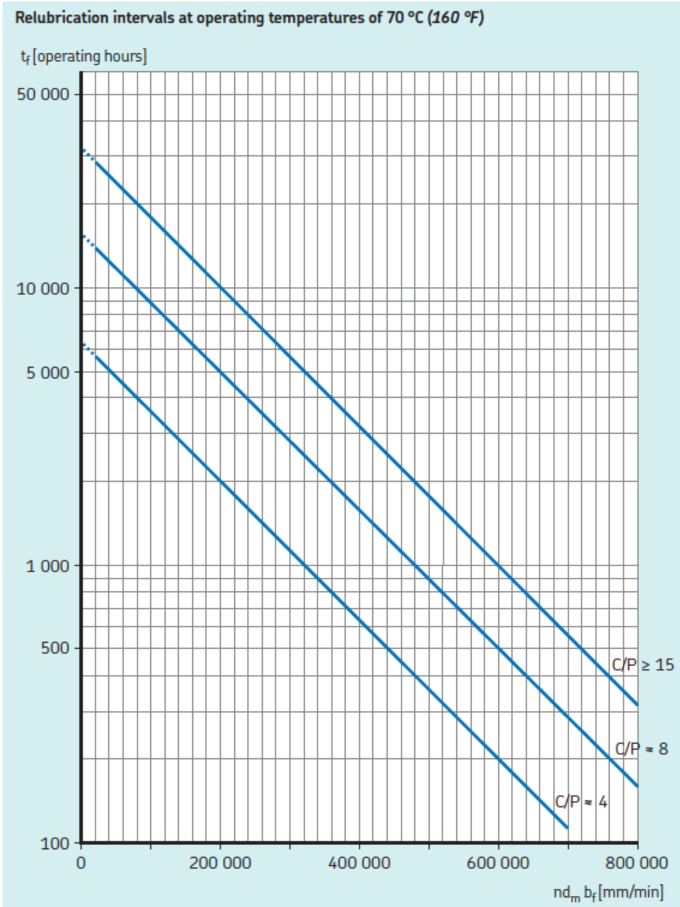


Figure 4.69: Rolling Bearing - SKF Relubrication Interval t_f [8]

Operating Temperature and Speed

As reported previously, the steps here reported, are not to be considered as a straight sequence, but as interdependent. Temperature and Speed, are two parameters with a strong relationship with the different steps already explained. An interactive approach to the analysis, is requested, in order to achieve an optimum design for a bearing arrangement and select the most appropriate components for it.
The operating temperature of a bearing is the steady-state temperature it attains when running and in thermal equilibrium with its surrounding elements. The operating temperature is affected by:

- The heat generated by the bearing, as a result of the combined bearing and seal frictional power loss. Friction is function of speed and lubricant selection, as shown in figure.

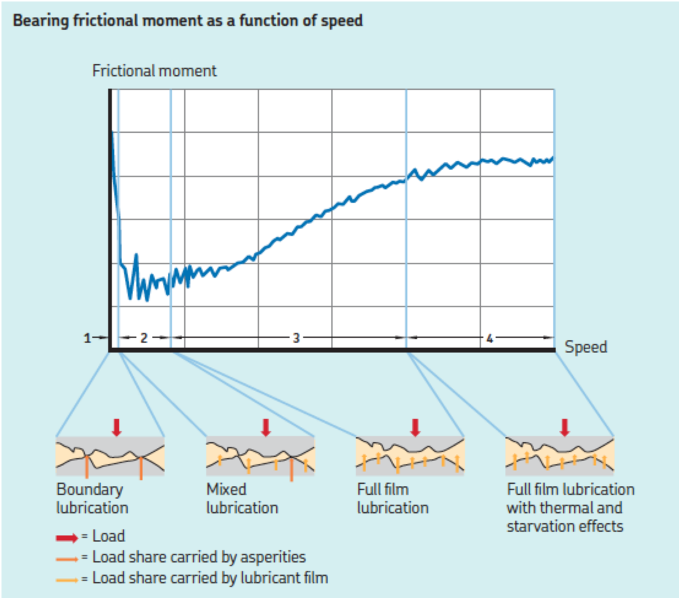


Figure 4.70: Rolling Bearing - Frictional Moment [8]

The equation to evaluate Total Frictional Moment

$$M = M_{rr} + M_{sl} + M_{seal} + M_{drag} \tag{4.28}$$

$$\begin{cases} M_{rr} = \text{Rolling Frictional Moment } [Nmm] \\ M_{sl} = \text{Sliding Frictional Moment } [Nmm] \\ M_{seal} = \text{Seal Frictional Moment} \\ M_{drag} = \text{Frictional Moment from Drag Losses} \end{cases}$$

And the Power Loss in $[W]$ results

$$P_{loss} = 1.05\ 10^{-4} M n \tag{4.29}$$

- The heat from the application transferred to the bearing via the shaft, housing, foundation and other elements.
- The heat dissipated from the bearing via the shaft, housing, foundation, lubricant cooling system (if used) and other cooling devices.

If can be estimated a value for the heat dissipation W_s, can be also defined the operating temperature, T_{bear}, for a bearing in thermal equilibrium, under steady-state conditions.

$$T_{bear} = \left(\frac{P_{loss}}{W_s}\right) + T_{amb} \tag{4.30}$$

The speed capability of a bearing is normally determined by the bearing operating temperature. However, for certain bearing types and arrangements, the mechanical limits of the bearing components may have a significant influence. The product tables typically provide two speed ratings: reference speed (based on thermal conditions) and limiting speed (based on mechanical limits).[8] Is advisable to not apply on bearings, speeds over the limits reported on product tables, temperature and mechanical properties could be strongly influenced, and the reliability of the bearings not guaranteed.

Bearing Interfaces

Bearing seats on shafts and in housings, and components which locate a bearing axially, have a significant impact on bearing performance. To fully exploit the load carrying ability of a bearing, its rings or washers should be fully supported around their complete circumference and across the entire width of the raceway. Bearing seats should be manufactured to adequate geometrical and dimensional tolerances and be uninterrupted by grooves, holes or other features.[8]
Fits for rolling bearings are typically specified with standard tolerance classes for holes and shafts as described in ISO 286-2. As bearings are typically manufactured to ISO tolerances, the selection of the tolerance class for the bearing seat determines the fit. The position and width of the tolerance intervals, of commonly used tolerance classes, relative to the bearing bore and outside diameter tolerances, are illustrated in fig. 4.71.[8]

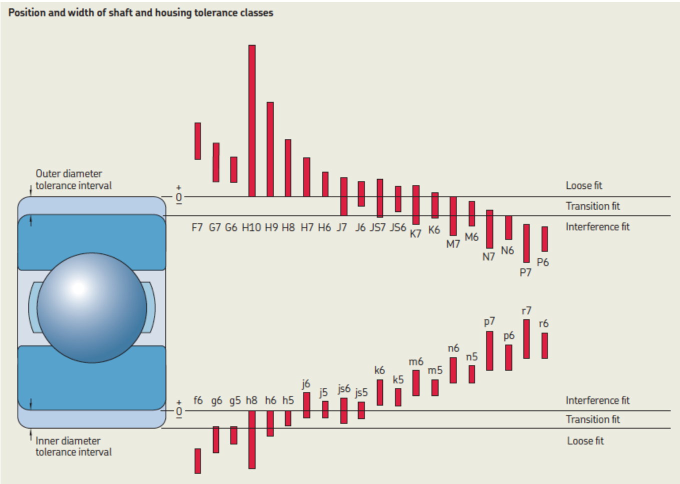


Figure 4.71: Rolling Bearing - Tolerance Classes [8]

The rotation conditions affect the performances of the bearing, due to the relative motion between bearing ring and load acting. The possible rotation conditions are:

- Rotating Loads: occur where the bearing ring or the applied load is stationary, while the other rotates. An adequate interference fit is requested between ring and seat, to avoid failure phenomena.
- Stationary Loads: occur where both the bearing ring and the applied load are stationary or rotating at the same speed. In this condition no interference fit requested.
- Load direction undetermined: variable or alternating external loads, sudden load peaks, vibration or unbalanced loads in high-speed applications. These give rise to changes in the direction of load, which cannot be accurately described. In this condition is recommended an interference fit for both rings.

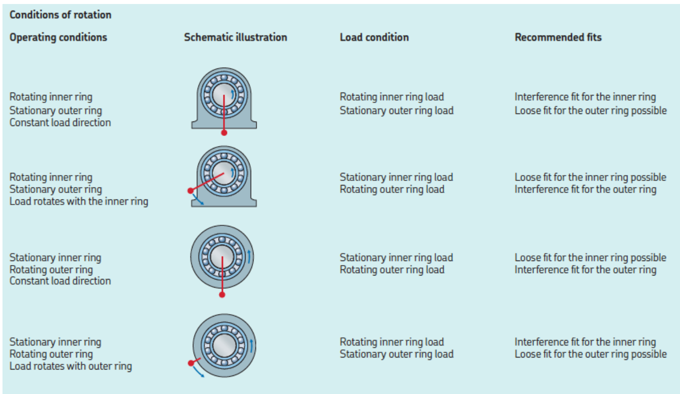


Figure 4.72: Rolling Bearing - Rotation Conditions [8]

The ring of a bearing deforms proportionately to the load. When exploiting a rotating inner ring, the deformation can affects the interference between inner ring and shaft. The heavier the load, the tighter is the interference Δ required.

$$\Delta = 2.5\sqrt{F_r \frac{d}{B}} \tag{4.31}$$

$$\text{With} \begin{cases} F_r = \text{Radial bearing load } [KN] \\ \Delta = \text{Required interference } [\mu m] \\ d = \text{Bore diameter } [mm] \\ B = \text{Bearing width } [mm] \end{cases}$$

Another parameter to take into account is the temperature. During operation, bearing rings reach higher temperatures than that of the surrounding components. This

can loosen the fit on the shaft seat, while outer ring expansion can prevent the desired axial displacement in the housing.
Hollow shafts and tapered seats require a specific design procedure, illustrated in [8], not reported here.
The following tables from SKF catalogue, provide recommendations for tolerances of shaft and housing seats They are valid for standard applications (only the tables of interest for this project are reported).

Tolerances for solid steel shafts – seats for radial ball bearings[1)]

Conditions	Shaft diameter	Dimensional tolerance[2)]	Total radial run-out tolerance[3)]	Total axial run-out tolerance[3)]	Ra
	mm	–	–	–	µm
Rotating inner ring load or direction of load indeterminate					
Light loads (P ≤ 0,05 C)	≤ 17	js5	IT4/2	IT4	0,4
	> 17 to 100	j6	IT5/2	IT5	0,8
	> 100 to 140	k6	IT5/2	IT5	1,6
Normal to heavy loads (0,05 C < P ≤ 0,1 C)	≤ 10	js5	IT4/2	IT4	0,4
	> 10 to 17	j5	IT4/2	IT4	0,4
	> 17 to 100	k5	IT4/2	IT4	0,8
	> 100 to 140	m5	IT4/2	IT4	0,8
	> 140 to 200	m6	IT5/2	IT5	1,6
	> 200 to 500	n6	IT5/2	IT5	1,6
	> 500	p7	IT6/2	IT6	3,2
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable		g6[4)]	IT5/2	IT5	1,6
Easy axial displacement of inner ring on shaft unnecessary		h6	IT5/2	IT5	1,6
Axial loads only		j6	IT5/2	IT5	1,6

Figure 4.73: Rolling Bearing - Seat Tolerances [8]

Tolerances for solid steel shafts – seats for radial roller bearings[1)]

Conditions	Shaft diameter	Dimensional tolerance[2)]	Total radial run-out tolerance[3)]	Total axial run-out tolerance[3)]	Ra
	mm	–	–	–	µm
Rotating inner ring load or direction of load indeterminate					
Light loads (P ≤ 0,05 C)	≤ 25	j6	IT5/2	IT5	0,8
	> 25 to 60	k6	IT5/2	IT5	0,8
	> 60 to 140	m6	IT5/2	IT5	0,8
Normal to heavy loads (0,05 C < P ≤ 0,1 C)	≤ 30	k6	IT5/2	IT5	0,8
	> 30 to 50	m5	IT5/2	IT5	0,8
	> 50 to 65	n5	IT5/2	IT5	0,8
	> 65 to 100	n6	IT5/2	IT5	0,8
	> 100 to 280	p6	IT5/2	IT5	1,6
	> 280 to 500	r6	IT5/2	IT5	1,6
	> 500	r7	IT6/2	IT6	3,2
Heavy to very heavy loads and high peak loads under difficult operating conditions (P > 0,1 C)	> 50 to 65	n5	IT5/2	IT5	0,8
	> 65 to 85	n6	IT5/2	IT5	0,8
	> 85 to 140	p6	IT5/2	IT5	0,8
	> 140 to 300	r6	IT5/2	IT5	1,6
	> 300 to 500	r6 + IT6[4)]	IT5/2	IT5	1,6
	> 500	r7 + IT7[4)]	IT6/2	IT6	3,2
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable		g6[5)]	IT5/2	IT5	1,6
Easy axial displacement of inner ring on shaft unnecessary		h6	IT5/2	IT5	1,6
Axial loads only		j6	IT5/2	IT5	1,6

Figure 4.74: Rolling Bearing - Seat Tolerances [8]

On SKF - Rolling Bearing-17000 EN are reported tables which provide information about bearing tolerances, seat tolerances and resultant fits. These enable to determine easily the maximum and minimum values of its when using ISO tolerance

classes for bearing seats and bearings with Normal tolerances for the bore and outside diameter.
Typically, it is not sufficient to use an interference alone to axially locate a bearing ring on a cylindrical seat. Common ways of locating bearing rings axially include:

- Shaft or housing shoulders.
- Lock nuts or threaded rings.
- End plates or housing covers.
- Distance rings, which support against adjacent parts.
- Snap rings.

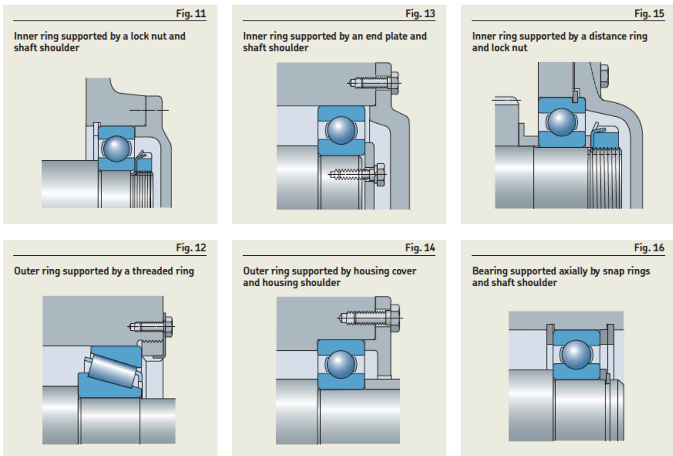


Figure 4.75: Rolling Bearing - Axial Location [8]

Bearing Execution

When bearing type, size and fit have been defined, additional factors must be considered.

- Selecting Clearance: bearing clearance is defined by SKF as the total distance through which one bearing ring can be moved relative to the other in the radial direction (radial internal clearance) or in the axial direction. Three different conditions, exposed in figure 4.76, are possible.

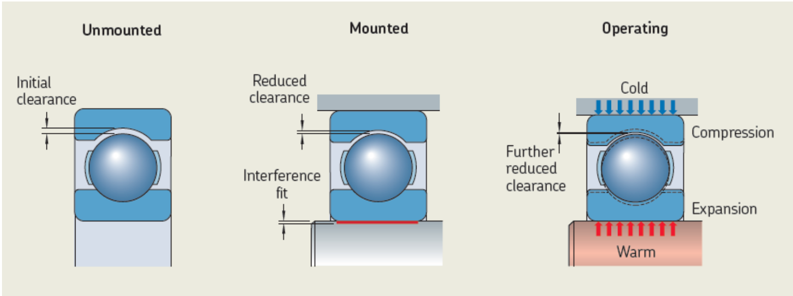


Figure 4.76: Rolling Bearing - Bearing Clearance [8]

Considering the Operational clearance (internal clearance in bearing when in operation and at stable temperature), or the pre-load in a bearing, their presence influence: the friction, load zone size and fatigue life of a bearing (fig.4.77).

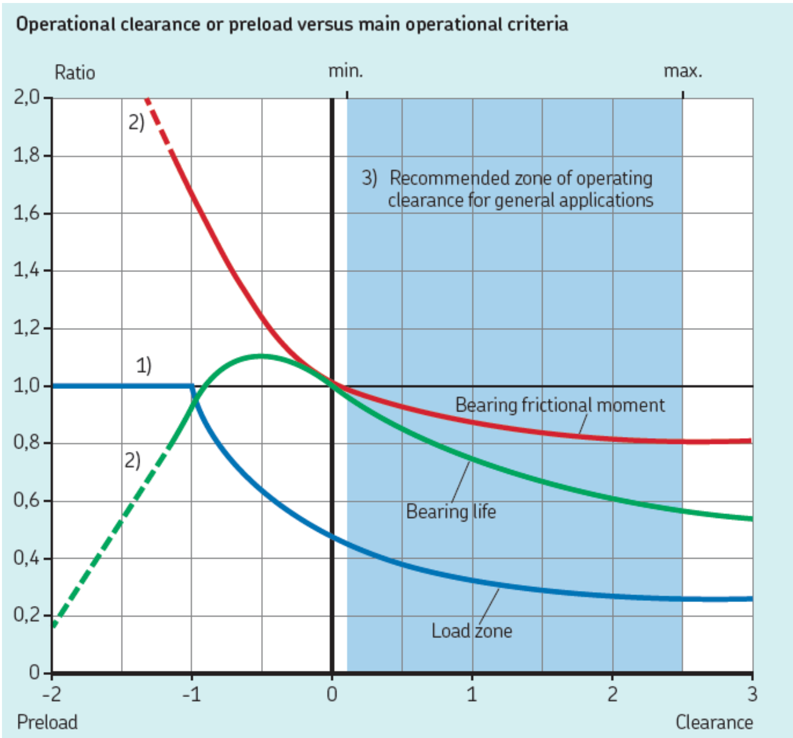


Figure 4.77: Rolling Bearing - Operational Clearance Effects [8]

For general applications, the operating clearance range, in rolling radial bearings, should be within the recommended blue zone in fig. 4.77. The resulting Operational clearance is affected by the choice of Initial clearance (internal clearance in the bearing prior to mounting). The objective, is ensure a minimum initial internal clearance, with a value that, when reduced by mounting effects (interference fits, temperature difference), is equal to or greater than the required minimum operating clearance. Initial clearance is determined during manufacturing, ISO has defined five clearance classes to specify the degree of initial internal clearance in a bearing. Each clearance class represents a range of values, which varies depending on bearing type and size. Clearance reduction caused by mounting is defined as

$$\Delta r_{fit} = \Delta_1 f_1 + \Delta_2 f_2 \tag{4.32}$$

$$\text{With} \begin{cases} \Delta_1 = \text{Effective interference inner ring-shaft } [\mu m] \\ \Delta_2 = \text{Effective interference outer ring-housing } [\mu m] \\ f_1 = \text{Reduction factor of inner ring} \\ f_2 = \text{Reduction factor of outer ring} \end{cases}$$

The clearance variation, function of the temperature is defined as

$$\Delta r_{temp} = 0.0012 \Delta T d_m \tag{4.33}$$

$$\text{With} \begin{cases} \Delta T = \text{Temperature difference inner-outer ring } [^\circ C] \\ d_m = \text{Mean bearing diameter } [mm] \end{cases}$$

The required minimum initial internal clearance can be estimated using

$$r = r_{op} + \Delta r_{fit} + \Delta r_{temp} + \Delta r_{other} \tag{4.34}$$

$$\text{With} \begin{cases} r_{op} = \text{Required operating clearance } [\mu m] \\ \Delta r_{fit} = \text{Clearance variation due to maximum expected fit } [\mu m] \\ \Delta r_{temp} = \text{Clearance variation due to } \Delta T \ [\mu m] \\ \Delta r_{other} = \text{Clearance variation due to other effects } [\mu m] \end{cases}$$

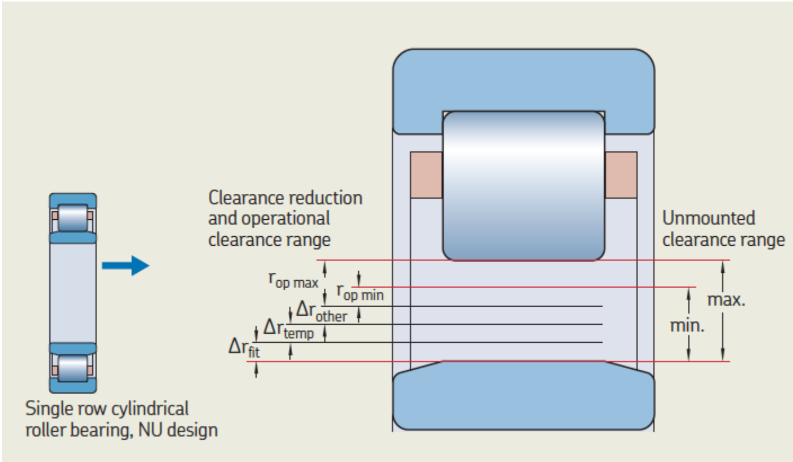


Figure 4.78: Rolling Bearing - Clearance Contributions [8]

- Selecting Pre-load: depending on the application, bearings may need a pre-load. When high degree of stiffness or positional control is required, then pre-load may be suitable. Similarly, where there is a very light or no external load, then pre-load may be required to ensure a minimum load.

Sealing, Mounting and Dismounting

Seals for bearing applications should provide maximum protection with a minimum amount of friction and wear, under the prevailing operating conditions. Because bearing performance and service life are so closely tied to the effectiveness and cleanliness of the lubricant, the seal is a key component. Many factors must be considered when selecting the most suitable seal for a particular bearing-shaft-housing system[8]:

- Lubricant type: oil or grease
- Contaminant type: particles or fluid
- Circumferential speed at the seal lip
- Shaft arrangement: horizontal or vertical
- Possible shaft misalignment or deflection
- Run-out and concentricity
- Available space

- Seal friction and the resulting temperature increase
- Environmental influences
- Cost
- Required operating time
- Maintenance requirements

The purpose of a seal is to retain lubricant and prevent any contaminants from entering into a controlled environment. There are three seals types:

- Non-contact seals
- Contact seals
- Static seals

Some technical solution example are reported here. For a complete overview of available sealing system, their performances and functioning, is recommended SKF - Rolling Bearing-17000 EN.

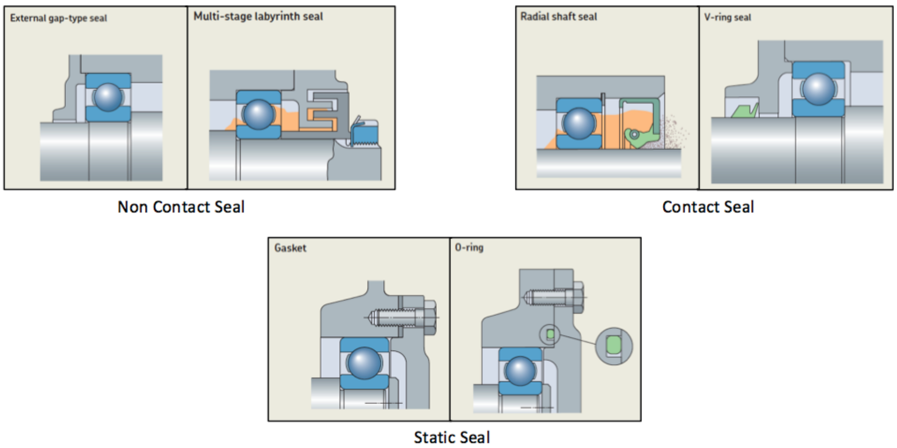


Figure 4.79: Rolling Bearing - Seal Type [8]

Rolling bearings are reliable machine elements that can provide long service life, if they are mounted properly. Proper mounting requires experience, accuracy, a clean work environment, correct working methods and the appropriate tools.
In this project of thesis, the size of bearings adopted, will not require tools to enliven them. In mounting and dismounting phases only the following items need to be respected:

- Determine the correct sequence and orientation in which components are to be assembled.

- Leave the bearings in their original packages until immediately before mounting, to avoid any contamination.

- Bearings should be mounted in a dry, dust free area.

- Check the cleaning of housings, shafts, seals and other components of the bearing-shaft-housing system.

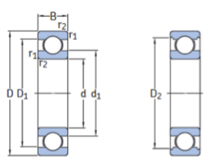
- Check the dimensional and geometrical tolerances of each component.

- Never apply the mounting force through the rolling elements (if possible adopt a Bearing fitting tool).

- For an interference it, the mating surfaces should be coated with a thin layer of light oil.

- Never apply the dismounting force through the rolling elements (if possible adopt a Mechanical puller).

SKF offers a tool to perform the calculation and validation of bearings, defining application conditions, *SKF Bearing Select*, based on bearing's theory reported previously. Firstly is selected a *Single row deep groove ball bearings*, designed to support radial load , which mainly affects the system. It's also available a solution with filling slots in both the inner and outer rings, to accommodate more balls, have a higher radial load carrying capacity. However their axial load carrying capacity is limited. From SKF catalogue is selected Deep groove ball bearing 16101, its characteristics are reported in figure 4.80.



BEARING PROPERTIES

	Designation SKF Explorer ▶ Popular item	Bearing type	Principal dimensions Bore d	 Outer diameter D	 width B	Basic load ratings Dynamic C	 Static C_0	Fatigue load limit P_u	Speed ratings Reference n_{ref}	 Limiting n_{lim}
			mm			kN			r/min	
Left	16101	Deep groove ball bearing	12	30	8	5.07	2.36	0.1	60000	38000
Right	16101	Deep groove ball bearing	12	30	8	5.07	2.36	0.1	60000	38000

Figure 4.80: Deep Groove Ball Bearing 16101

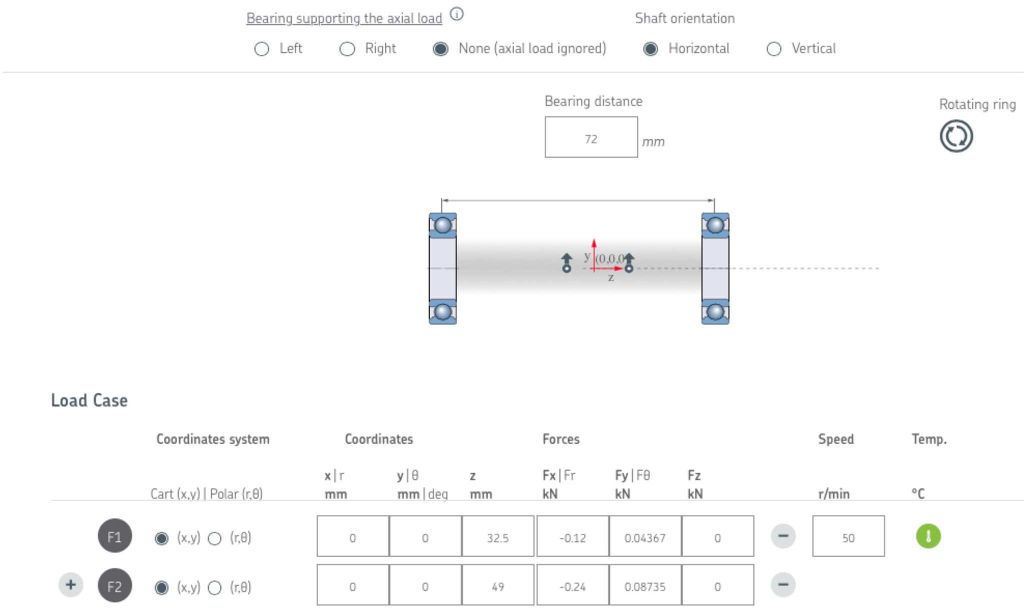


Figure 4.81: Deep Groove Ball Bearing - SKF Bearing Select

In 4.81 are visible the working conditions adopted, which are extrapolated from previous calculations in section 4.5.2: Force module, Force directions, Force origin, Bearings relative position, Rotational speed, Working temperature.
The results, obtained from calculations, are reported in following table, bearings are differentiated depending on their positioning respect rack-pinion systems. The bearings selected, for shift system, satisfy the required performances. The permissible operating temperature depends on material selected for seals:

- NBR:$[-40 \div +100]°C$, up to $120°C$ can be tolerated for brief periods.

- FKM: $[-30 \div +200]°C$, up to $230°C$ can be tolerated for brief periods.

FKM seals are recommended to satisfy requirements from any customer.

Table 4.22: Deep Groove Ball Bearing - SKF Bearing Select

Item	$Shift_{out}$ Side	$Shift_{in}$ Side
Equivalent Dynamic Load P $[kN]$	0.15	0.23
Equivalent Static Load P_0 $[kN]$	0.152	0.231
Static Safety Factor s_0	15.6	10.2
Load Ratio C/P	33.43	21.9
Radial Reaction Force F_r $[kN]$	0.152	0.231
Actual Operating Viscosity ν $[mm^2/s]$	11	11
Rated Operating Viscosity ν_1 $[mm^2/s]$	362	362
Actual Operating Viscosity $(40°C)$ ν_{ref} $[mm^2/s]$	> 1000	> 1000
Viscosity Ratio κ	0.03	0.03
Grease Relubrication Interval t_f $[h]$	5960	5960
Speed Factor nd_m $[mm/min]$	1000	1000
Adjusted Reference Speed n_{ar} $[r/min]$	58900	56500
Adjustment Factor for P f_p	0.98	0.94
Adjustment Factor for oil viscosity f_v	1	1
Inner Ring Rotational Frequency f_i $[Hz]$	0.83	0.83
Outer Ring Rotational Frequency f_e $[Hz]$	0	0
Rolling Element Rotational Frequency f_r $[Hz]$	0.31	0.31
Inner Ring Over-rolling Frequency f_{ip} $[Hz]$	4.12	4.12
Outer Ring Over-rolling Frequency f_{ep} $[Hz]$	2.53	2.53
Rolling Element Over-rolling Frequency f_{rp} $[Hz]$	3.3	3.3
Frictional Moment M $[Nmm]$	1.22	2.36
Frictional Moment at Start M_{start} $[Nmm]$	1.35	2.74
Rolling Frictional Moment M_{rr} $[Nmm]$	0.14	0.18
Sliding Frictional Moment M_{sl} $[Nmm]$	1.08	2.18
Seals Frictional Moment M_{seal} $[Nmm]$	0	0
Drag Frictional Moment M_{drag} $[Nmm]$	0	0

The same procedure is adopted to verify bearings on select system. The validation is carried out on same kind of bearings, Deep groove ball bearing 16101 (fig. 4.82). In table 4.23 are reported the values form calculations. Even for select application the Bearing 16101, results to satisfy the required performances.

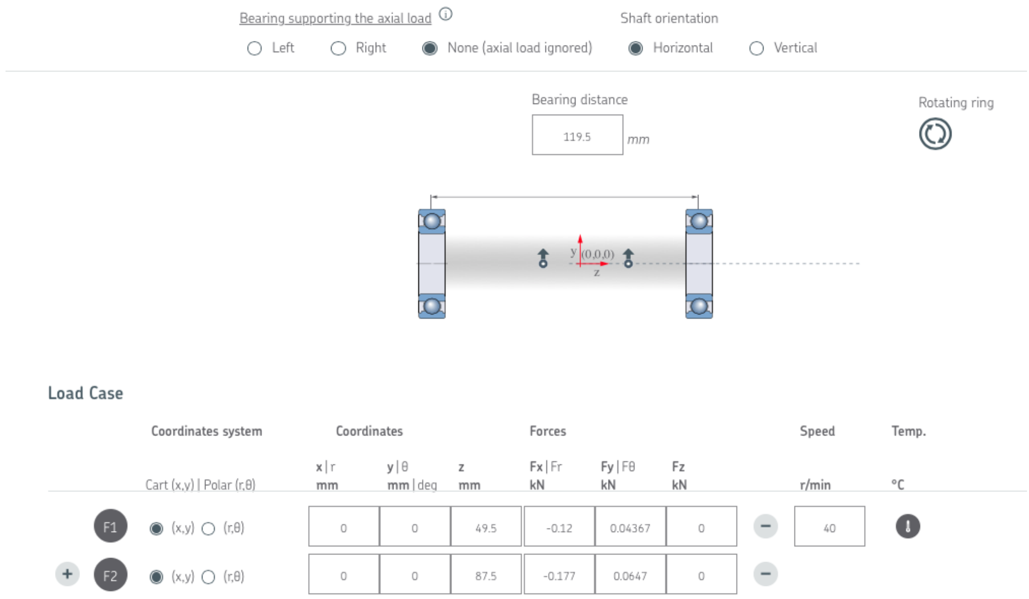


Figure 4.82: Deep Groove Ball Bearing - SKF Bearing Select

Table 4.23: Deep Groove Ball Bearing - SKF Bearing Select

Item	$Select_{out}$ Side	$Select_{in}$ Side
Equivalent Dynamic Load P $[kN]$	0.13	0.19
Equivalent Static Load P_0 $[kN]$	0.125	0.191
Static Safety Factor s_0	18.8	12.4
Load Ratio C/P	40.44	26.54
Radial Reaction Force F_r $[kN]$	0.125	0.191
Actual Operating Viscosity ν $[mm^2/s]$	11	11
Rated Operating Viscosity ν_1 $[mm^2/s]$	439	439
Actual Operating Viscosity (40°C) ν_{ref} $[mm^2/s]$	> 1000	> 1000
Viscosity Ratio κ	0.02	0.02
Grease Relubrication Interval t_f $[h]$	5970	5970
Speed Factor nd_m $[mm/min]$	800	800
Adjusted Reference Speed n_{ar} $[r/min]$	59700	57700
Adjustment Factor for P f_p	1	0.96
Adjustment Factor for oil viscosity f_v	1	1
Inner Ring Rotational Frequency f_i $[Hz]$	0.66	0.66
Outer Ring Rotational Frequency f_e $[Hz]$	0	0
Rolling Element Rotational Frequency f_r $[Hz]$	0.25	0.25
Inner Ring Over-rolling Frequency f_{ip} $[Hz]$	3.3	3.3
Outer Ring Over-rolling Frequency f_{ep} $[Hz]$	2.03	2.03
Rolling Element Over-rolling Frequency f_{rp} $[Hz]$	2.64	2.64
Frictional Moment M $[Nmm]$	0.9	1.73
Frictional Moment at Start M_{start} $[Nmm]$	0.98	1.99
Rolling Frictional Moment M_{rr} $[Nmm]$	0.11	0.14
Sliding Frictional Moment M_{sl} $[Nmm]$	0.78	1.59
Seals Frictional Moment M_{seal} $[Nmm]$	0	0
Drag Frictional Moment M_{drag} $[Nmm]$	0	0

The selection of Linear Bearings in made through SKF Linear Bearing and Units - Technical Handbook[9].
The selection of suitable Linear Ball Bearing follows two main steps:

- Life calculation (under dynamic operating conditions): distance covered between the guidance elements before the first sign of material fatigue occurs.
- Static safety load calculation (under static operating conditions): ratio between the basic static load rating and the static equivalent load, gives the degree of safety against excessive permanent deformation of the rolling elements and raceways.

Here are reported the main concepts, useful to select properly a linear bearing, for this application, to obtain a complete overview of the parameters affecting linear bearing performances, consult SKF Linear Bearing and Units - Technical Handbook[9].

- Life of Linear Ball Bearing: The life of a linear ball bearing is defined as the distance travelled (or the number of operating hours/strokes at constant stroke length and frequency) by the bearing before the first sign of material fatigue (spalling) appears on the raceway or rolling elements.
  All references to the dynamic load rating of SKF linear ball bearings apply to the basic rating life, as covered by the ISO definition, in which the life is understood as that reached or exceeded by 90% of a large group of identical bearings. The majority of the bearings reach a longer life and half the total number of bearings reach five times the basic rating life.[9]
- Service Life: The service life is the period of time for which a given linear bearing remains operational in a given set of operating conditions. The service life of a bearing therefore depends not necessarily on fatigue but also on wear, corrosion, seal failure, lubrication intervals.[9]
- Basic Rating Life L: The basic rating life is the life that 90% of a sufficiently large group of apparently identical linear rolling bearing can be expected to attain or exceed under identical operating conditions.
- Total Life L_{ges}: The total life for i identical rolling elements with the dynamic load rating C is obtained from

  $$L_{ges} = \frac{C^p}{\left[\Sigma P_i^{(\beta p)}\right]^{(1/\beta)}} \tag{4.35}$$

  Exponent β of the two-parameter Weibull distribution, is usually set at 10/9, while p is the life exponent ($p = 3$ for ball bearings, $p = 10/3$ for roller bearings).

- Basic Dynamic Load Rating C: The basic dynamic load rating is the radial load, constant in magnitude and direction, which a linear rolling bearing can theoretically bear for a basic rating life of 100000 m of travel. The basic rating life L, is influenced by a short stroke, taken into consideration with factor f_s.

$$L = f_s \left(\frac{C}{P} \right)^P \tag{4.36}$$

The factor f_s is defined from table 4.83 as the ratio of the single stroke l_s and the support length l_t of the rolling element.

l_S / l_t	$f_{S,ball}$
1,0	1,00
0,9	0,91
0,8	0,82
0,7	0,73
0,6	0,63
0,5	0,54
0,4	0,44
0,3	0,34
0,2	0,23
0,1	0,13

Figure 4.83: Linear Ball Bearing - Factor f_s [9]

- Effective Dynamic Load Rating C_{eff}: The dynamic load rating values given in the SKF linear rolling bearing tables are valid for a direction of load which correspond to the maximum load carrying capacity of the bearings operating under optimum conditions. To take into account operating conditions which differ from this optimum, it is necessary to modify the basic dynamic load rating by a number of factors to give an effective dynamic load rating which is then inserted in the life equation.

$$C_{eff} = f_h f_i C \tag{4.37}$$

$$\text{With} \begin{cases} f_h \text{ Factor for surface hardness of shaft} \\ f_i = i^w \text{ Factor for number of loaded bearings} \\ (w = 0.7 \text{ ball bearings, } w = 7/9\text{roller bearings}) \end{cases}$$

Factor f_h is primarily applicable when softer-than-usual steel shafts are used with linear ball bearings. Steel shafts for linear guidance systems should, like

the raceways of linear ball bearings, be hardened and ground. The surface hardness should be at least 58 HRC and the mean surface roughness R_a measured to DIN 4768, should never exceed 0.32 μm. If shafts with a lower surface hardness are used, the factor f_h should be evaluated as $f_h = (HV/655)^2$, when calculating the effective dynamic load rating. Lower surface hardness also influences the basic static load rating C_0. Values shown in the catalogue should be corrected using the factor $f_{h0} = (HV/555) - 0.17$.
The equations reported above are valid for $HV = 400 \div 700$.
For $HV > 700$ the factors become equal to 1. [9]

- Permissible Dynamic Moments M_{max}: The dynamic moments $M_{x,max}$, $M_{y,max}$, $M_{z,max}$ complement the basic dynamic load rating C. As pure moments around the x, y or z axes they are parameters of the reliable dynamic load carrying capacity, in particular for profile rail guide carriages, but also for precision rail guide tables. Like the basic load rating C they relate to 100 km travel and a failure probability of 10%.[9]

- Equivalent Dynamic Bearing Load P: The equivalent dynamic bearing load is the constant radial load in magnitude and direction under the influence of which a linear ball bearing would reach the same basic rating life as under the actual load conditions. The equivalent dynamic bearing load should not exceed a value $C/2$ or the static load rating C_0. If the load F acting on the linear rolling bearings corresponds to the requirements for the basic load rating C, then $P = F$ and the load can be inserted directly into the life equation. In all other cases it is necessary to calculate the equivalent dynamic bearing load[9].

$$P = F f_t f_l f_m \tag{4.38}$$

With $\begin{cases} f_t = 1 + (t-100)^2/60000 \text{ Factor for operating temperature} \\ f_l \text{ Factor for direction of load } [N] \\ f_m \text{ Factor for misalignment} \end{cases}$

Linear rolling guides with metal cages and end pieces can usually be used at temperatures of up to 120°C; even higher temperatures up to 150°C are tolerable for brief periods.

- Mean Load F_m: The mean load is evaluated as

$$F_m = (l_F/l_V)1/p \tag{4.39}$$

With $\begin{cases} l_F = \int_0^T v(t) F^p(t) dt \\ l_V = \int_0^T v(t) \end{cases}$

As a general rule, the equivalent bearing load P should not exceed 50% of the dynamic load rating C: $P \leq C/2$

- Basic Static Load Rating C_0: The basic static load rating, is the static load which corresponds to a calculated load, applied through the centre of the highest loaded contact zone, between shaft and rolling elements of 5300 MPa. This stressing is expressed as the maximum Hertzian pressure which the linear rolling bearing can tolerate from experience. These values vary slightly for the different types of linear rolling bearings. The resulting total permanent deformation of rolling elements and raceway represents approximately 0.0001 of the rolling element diameter.[9]

- Static Load Carrying Capacity: When selecting a linear rolling bearing, the basic static load rating C_0 must be considered when in following cases:

  - The bearing is stationary and is loaded for long periods or is shock loaded.
  - The bearing operates under load at very low speeds.
  - The bearing operates normally but must also accept heavy shock loads.

  The permissible load is determined not through material fatigue, but through the permanent physical deformation at the contact zone of the rolling elements and raceways. Load applied causes flattening of the rolling elements and results in damage to the raceways. The damage may be uneven or may be spaced along the raceway at intervals corresponding to the rolling element separation. This permanent deformation leads to vibration in the bearing, noisy running and increased friction and may even cause an increase in clearance. With continued operation this permanent deformation may become a starting point for fatigue damage due to resulting peak loads. When determining the bearing size, according to static load carrying capacity, must consider a certain relationship, known as the static safety factor s_0, between the basic static load rating C_0 and the equivalent load P_0 in order to obtain the static load rating of the bearing.[9]

- Effective Static Load Rating $C_{0,eff}$: The static load ratings quoted in the SKF linear bearing tables, are valid for that direction of load conforming to the maximum load carrying capacity of the linear rolling bearing and where the bearing operates under optimum conditions. In order to take into consideration different operating conditions, in the static load safety equation, the effective static load rating needs to be calculated by including the most important operating factors.[9]

  $$C_{0,eff} = f_{l0} f_m f_{h0} C_0 \tag{4.40}$$

  These factors include the influence of the direction of load, misalignment, raceway hardness, number of bearings. However the effective static load rating is often replaced by the recommendation of a static load safety s_0 value adjusted to the operating conditions.[9]

- Static Load Safety s_0: The static load safety, expressed as the ratio between the basic static load rating and the equivalent static load, gives the degree of safety against excessive permanent deformation of the rolling elements and raceways. Depending on the operating conditions and requirements on the quietness of running, a static load safety s_0 according to figure 4.84 is recommended based on experience.[9]

Operating conditions	s_0 from	up to
smooth, vibration-free	1	2
normal running	2	4
shock loads or vibration	3	5

Figure 4.84: Linear Ball Bearing - Static Load Safety s_0 [9]

- Permissible Static Moments $M_{0,max}$: The maximum permissible static moments given in many linear rolling bearing catalogues correspond to a static load safety $s_0 = 1$.

- Equivalent Static Bearing Load P_0: The equivalent static bearing load is defined as that static load which, if applied, would cause the same permanent deformation in the bearing as the actual load. This is determined by the maximum load F_{max}, which can occur at any time.[9] P_0 is calculated approximately as follows:
$$P_0 = F_0 + f_{T0}(F_{0,max} + C_{0,eff} M/M_{0,max}) \tag{4.41}$$

- Life Calculation: The basic rating life of a linear rolling bearing may be calculated from:
$$L_{10} = f_S(C/P)^p \tag{4.42}$$
Where the stroke length and frequency are constant it useful to calculate the basic rating life in hours of operation or number of double strokes using the equations:
$$L_{10h} = 510^7 f_S \frac{(C/P)^p}{(60sn)} L_{10d} = 510^7 f_S \frac{(C/P)^p}{s} \tag{4.43}$$

- Minimum Load: In order to assure slip-free running of a linear rolling bearing, the load must be kept higher than a predefined minimum value. As a general guideline, a load of $P = 0.02C$ is acceptable. Minimum load is vital in linear guidance systems which operate at high speed or with high acceleration. In such cases the inertia forces of the rolling elements and the friction within the lubricant can have an adverse effect on the rolling conditions in the bearing and can lead to damaging slip conditions between the rolling elements and raceways.[9]

- Permissible Operating Temperature: The permissible operating temperature range for SKF linear rolling bearings is $[-20°C \div +80°C]$. It is dictated by the cage and seal materials and applies to continuous operation. Lower and higher temperatures can be tolerated for brief periods.

- Permissible speed and acceleration: Permissible speed and acceleration are both largely determined by the contact forces between the rolling elements and raceways. Under normal operating conditions, in particular under minimum loads, the permissible speed is 5 m/s and the permissible acceleration is 100 m/s^2. Higher running speeds and further acceleration are possible, depending on the bearing design, bearing size, applied load, lubricant and bearing pre-load.[9]

- Minimum stroke: Prolonged operation of the guide under short stroke and high frequency, lead to increased wear in the rolling contact of the raceways that result in corrosion of rolling element.[9]

- Stationary conditions: Damage can occur to linear rolling bearings where they are stationary for long periods and subject to vibration from external sources. Micro-movement in the contact zone between rolling elements and raceways can damage the surfaces. This will cause significant increase in running noise and premature failure through material fatigue. Damages of this kind through vibration should be avoided, by isolating the bearings from external vibration.[9]

- Friction: Friction in a linear guidance system is affected by type and size of the bearing, the operating speed, as well as the quality and quantity of the lubricant used. The cumulative running resistance of a linear rolling bearing is defined by the levels of several factors: the rolling and sliding friction at the rolling elements' contact zone, friction at the points of sliding contact between the rolling elements and cage and also friction at the guiding surfaces of the return zones.[9]

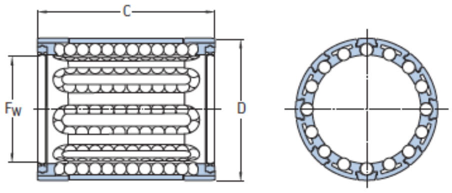
The following conditions are adopted to calculate the Load rating C and select the Linear Bearings:

- Distance between Select force application and Center of bearings $d = 60.5\ mm$

- Distance between Select force application and Knob interface $d = 92\ mm$

- Force applied to the lever $F = 100\ N$
- Force applied to Linear Bearings center $F_b = 152\ N$
- Force applied to single Linear Bearing $F_{bi} = 38\ N$
- The Speed of Shaft sliding inside the bearing $v = 115\ mm/s = 0.115\ m/s$

With the parameters reported above, an alternative solution to Linear Ball Bearings is considered, Linear Plain Bearings.

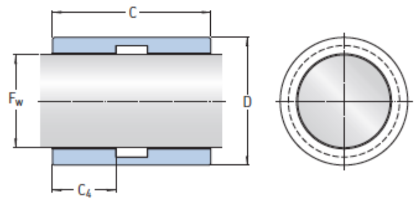


LBBR with double lip seals

Dimensions			No. of ball rows	Basic load ratings		Mass	Designations			
							Linear ball bearings		stainless steel	
F_w	D	C		dyn. C	stat. C_0		standard design	with 2 double lip seals	standard design	with 2 double lip seals
mm			—	N		kg	—			
3	7	10	4	60	44	0,0007	LBBR 3[2]	LBBR 3-2LS[2]	LBBR 3/HV6[2]	LBBR 3-2LS/HV6[2]
4	8	12	4	75	60	0,001	LBBR 4[2]	LBBR 4-2LS[2]	LBBR 4/HV6[2]	LBBR 4-2LS/HV6[2]
5	10	15	4	170	129	0,002	LBBR 5[2]	LBBR 5-2LS[2]	LBBR 5/HV6[2]	LBBR 5-2LS/HV6[2]
6	12	22[1]	4	335	270	0,006	LBBR 6A	LBBR 6A-2LS	LBBR 6A/HV6	LBBR 6A-2LS/HV6
8	15	24	4	490	355	0,007	LBBR 8	LBBR 8-2LS	LBBR 8/HV6	LBBR 8-2LS/HV6
10	17	26	5	585	415	0,011	LBBR 10	LBBR 10-2LS	LBBR 10/HV6	LBBR 10-2LS/HV6
12	19	28	5	695	510	0,012	LBBR 12	LBBR 12-2LS	LBBR 12/HV6	LBBR 12-2LS/HV6
14	21	28	5	710	530	0,013	LBBR 14	LBBR 14-2LS	LBBR 14/HV6	LBBR 14-2LS/HV6
16	24	30	5	930	630	0,018	LBBR 16	LBBR 16-2LS	LBBR 16/HV6	LBBR 16-2LS/HV6
20	28	30	6	1 160	800	0,021	LBBR 20	LBBR 20-2LS	LBBR 20/HV6	LBBR 20-2LS/HV6
25	35	40	7	2 120	1 560	0,047	LBBR 25	LBBR 25-2LS	LBBR 25/HV6	LBBR 25-2LS/HV6
30	40	50	8	3 150	2 700	0,070	LBBR 30	LBBR 30-2LS	LBBR 30/HV6	LBBR 30-2LS/HV6
40	52	60	8	5 500	4 500	0,130	LBBR 40	LBBR 40-2LS	LBBR 40/HV6	LBBR 40-2LS/HV6
50	62	70	9	6 950	6 300	0,18	LBBR 50	LBBR 50-2LS	LBBR 50/HV6	LBBR 50-2LS/HV6

Figure 4.85: Linear Ball Bearing - Selection

These bearings can be used in certain applications, where the use of rolling element bearings is inappropriate, because of different operating conditions. This is especially relevant in cases of heavy shock loads, vibration or where high speeds and acceleration are required under light load conditions. For such applications dry sliding bearings are preferable to linear ball bearings.[9]
The suitability of linear plain bearings for a given application depends largely on friction, heat dissipation, sliding properties of the mating surfaces and the efficiency of lubrication. [9]
The Basic dynamic load rating C of alinear plain bearing, represents the magnitude



LPBR

Dimensions				Basic load ratings			Mass	Designation
				dyn. at		stat.		Linear plain bearing
				0,1 m/s	4 m/s			
F_w	D $_{-0,07}$	C	C_4	C	C	C_0		
mm				N			kg	—
12	19,19	28	10	965	24	3 350	0,006	**LPBR 12**
14	21,21	28	12	1 370	34	4 750	0,007	**LPBR 14**
16	24,23	30	12	1 530	38	5 400	0,009	**LPBR 16**
20	28,24	30	13	2 080	52	7 350	0,011	**LPBR 20**
25	35,25	40	17	3 400	85	12 000	0,024	**LPBR 25**
30	40,27	50	20	4 800	120	17 000	0,033	**LPBR 30**
40	52,32	60	24	7 650	193	27 000	0,063	**LPBR 40**
50	62,35	70	27	10 800	270	38 000	0,088	**LPBR 50**

Figure 4.86: Linear Plain Bearing - Selection

and direction of a constant load, which influences its nominal service life under continuous linear movement at a given speed and at room temperature, expressed in running distance. The Basic static load rating C_0 is used for a linear plain bearing loaded when stationary (or in occasional slight adjustment movement). This gives an indication of the load which can be accepted by a linear plain bearing without exceeding a prescribed degree of distortion of the sliding surface.[9]
A fast check can be made to verify the proposed bearing under the specific operating conditions. If the initial conditions are below the permissible limits indicated in fig. 4.87, it can be assumed that the life of the bearing will be adequate. If however the maximum limits are exceeded, a larger size of bearing should be selected, in order to achieve the required pv value through reduction of the specific surface loading. The formulas to use the diagram are here reported

$$\begin{cases} v = \frac{sn}{30000} \\ p = \frac{P}{2F_wC_4} \end{cases} \tag{4.44}$$

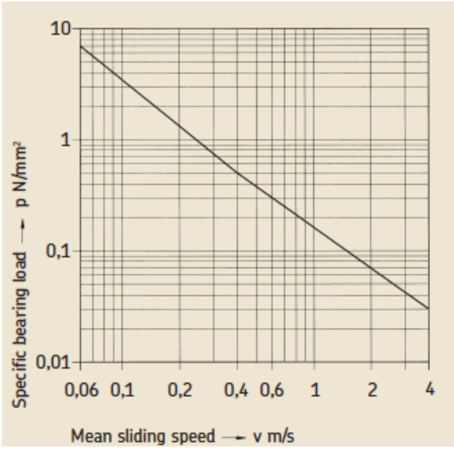


Figure 4.87: Linear Plain Bearing - pv Diagram

Selecting the Linear Plain Bearing LPBR12 the values obtained are $v = 0.1\ m/s$ and $p = 0.056\ N/mm^2$. The bearing selected results inside the limits in fig. 4.87, so is expected an adequate life for the component. Moreover the Basic load rating at the working speed is greater the the load acting on the bearing, so it can support even higher loads.
The sliding material of LPBR12 is polyacetal incorporating a layer of polyethylene. This combination, is particularly suitable for dry sliding bearing applications, and is characterised by its excellent resistance to wear. The maximum acceptable load rating is 14 N/mm^2. Recommended operating temperatures for continuous operation in the range $[-40 \div 80]°C$ and for short periods can support $120°C$.[9]

4.5.5 Dimensioning of Springs

To guarantee the correct functioning of the actuation system, springs dimensioned properly are necessary. The springs must support the weight of shift system and guarantee the contact of the pulley along the arch, for the complete stroke in select direction.
The following parameters are useful for the correct dimensioning of the springs:

- Components weight (Rack-Pinion systems 3 and 4, Linear plain bearings, Linear guides for racks, components 10,11,13, Pulley) $w = 3.4\ kg$.
- Mounting length allowed to springs $l_{spring,m} = 24\ mm$.
- Maximum solid height $l_{solid,max} = 18.8\ mm$.
- Minimum internal diameter: a clearance is required between spring and surrounding elements. Table 4.24 reports the suggested clearance depending on

spring size. Adopting a compression spring, must avoid an inner diameter too tight or loose around the shaft.

Table 4.24: Spring Recommended Clearance

Spring Diameter $[mm]$	Clearance $[mm]$
$0.762 \div 1.524$	0.254
$1.524 \div 7.92$	0.381
$7.92 \div 15.87$	0.508
$15.87 \div 25.4$	0.635
$25.4 \div 38.1$	0.762
$38.1 \div 50.8$	0.889
$50.8 \div 76.2$	1.016
$76.2 \div 101.6$	1.143

Considering the inner shaft with a diameter of 12 mm is selected the minimum internal diameter as $d_{min} = 13\ mm$.

- Maximum external diameter of spring $d_{max} = 20\ mm$.

As refernce for the selection of the spring is adopted Lee Spring Catalogue - 23 Series [10].

LEE STOCK NUMBER	OUTSIDE DIAMETER		TO WORK IN HOLE DIA. MIN		NOMINAL WIRE DIAMETER		TO WORK OVER ROD DIA. MAX		APPROXIMATE LOAD AT SOLID HEIGHT		NOMINAL FREE LENGTH		SPRING RATE		APPROXIMATE SOLID HEIGHT		PRICE GROUP		
																	Music Wire	302 Stainless	316 Stainless
	MM	IN	MM	IN	MM	IN	MM	IN	N	LB	MM	IN	N/MM	LB/IN	MM	IN	M	S	S316
LC 049HJ 01	16.76	0.660	17.45	0.687	1.24	0.049	13.67	0.538	44.48	10.00	15.88	0.625	4.08	23.30	4.62	0.182	J	L	Q
LC 049HJ 02											19.05	0.750	3.19	18.20	5.33	0.210	J	L	Q
LC 049HJ 03											22.23	0.875	2.68	15.30	5.84	0.230	K	M	R
LC 049HJ 04											25.40	1.000	2.31	13.20	6.32	0.249	K	M	R
LC 049HJ 05											31.75	1.250	1.80	10.30	7.32	0.288	K	M	R
LC 049HJ 06											38.10	1.500	1.49	8.50	8.33	0.328	L	N	S
LC 049HJ 07											44.45	1.750	1.26	7.20	9.32	0.367	L	N	S
LC 049HJ 08											50.80	2.000	1.10	6.30	10.31	0.406	M	P	T

Figure 4.88: Lee Spring Catalogue - 23 Series [10]

The spring *LC 049HJ 05*, underlined in figure 4.88, with end coils closed and ground square, results to satisfy the required performances. The inner diameter of the spring is $d_{in} = 14.28\ mm > d_{min}$ and the external $d_{ext} = 16.76\ mm < d_{max}$. When the spring is mounted, its length is $l_{spring,m} = 24\ mm$, the force generated by the spring in mounting position

$$F_{spring,m} = (l_{spring,free} - l_{spring,m})k = 13.95\ N$$

The system exploits 4 springs, the total force required is $F_{spring,tot} = w\ 9.81 = 33.35\ N$, so the springs are able to sustain the weight and generate an additional force, to guarantee the contact of the pulley along the arch. The delta between mounting and solid height of the spring, allows the complete stroke in z direction of the shift system $\Delta_{spring} = l_{spring,m} - l_{solid} = 16.68\ mm < l_{solid,max}$.

4.5.6 Design of components

Once dimensioned the main elements, and positioned them in the space x,y,z respect to the gearshift system, is necessary design all the components to link the parts and allow their correct functioning. This step hasn't a fixed position in the design process, it is a cardinal point to match correctly all the components, evaluate the packaging and define technical solutions, to reach the final realization of the system. The thickness of the elements and their geometry is continuously influenced and modified by the development of the system. This step transform the concept idea behind the solution developed, and interface it with the practical solutions required for the correct mounting and functioning of the project, examples: position and dimensions of the screws, mounting of the actuation cables, bearings seats and locking solutions.

4.5.7 Adaptation of Actuation system to Test Bench layout

This step is strictly linked to the previous one. The actuation system is integrated on a test bench, where the vehicle layout is reproduced. This integration, requires the design of components to position the system on the bench at the correct coordinates and with reliable solutions. The system must be fixed without interfering with existing elements. To do so, is necessary change the reference system and adopting a new one referred to the test bench. In this step are developed all the components marked as 1 in chapter 4.5. As underlined above, the design of those components has a direct interdependence with the previous step, as all the elements interface and contribute to the robustness of the system. Both this two final steps, require a dimensioning of the the components from the structural point of view. As the objective of this project of thesis is the development of an actuation system and a Resistance system, the structural validation of the designed components is not performed, it will require a deep study and a validation through CAE software. The thickness and geometries of the elements are designed using the experience.

Chapter 5

Test Bench Reaction Design

5.1 Reaction System - Solution 1

The first solution, for a reaction system with active resistance, exploits pneumatic force. The concept, exposed in figure 5.1, is to adopt a double effect pneumatic cylinder to generate reaction forces against cable end motion, with the piston actuated by compressed air. The force generated by the cylinder is directly proportional to

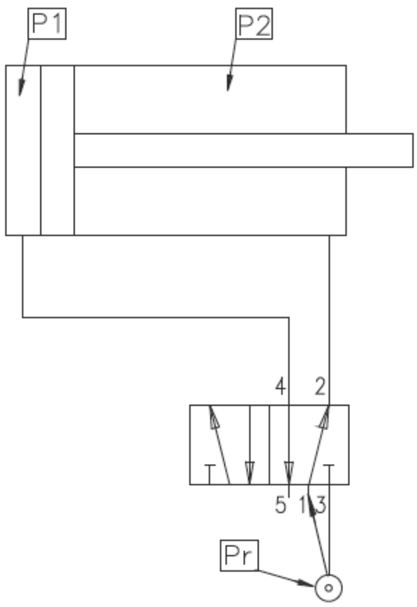


Figure 5.1: Solution 1 - Concept[11]

the source pressure and to piston surface $F_c = pS$, on cable's side the surface S is reduced by piston's rod area s. The force generated by the cylinder, is influenced by friction losses generated by seals, for this reason the real force is reduced about $10 - 15\%$. The sliding speed is influenced by:

- Internal friction.
- Load applied.
- Cylinder Mounting positioning.
- Air Flow.

With the correct regulation, is possible achieve the desired speed. In fig. 5.2 can be seen the components of a double effect cylinder. On the piston is mounted a end-stop, which function is to smooth the end stroke, it slowly increases the force in the last part of the motion (fig. 5.3). The end-stop reduces the kinetic energy of the piston, to avoid impacts at the end stroke, which would compromise the functionality of the cylinder. Cylinders without end-stop aren't recommended in high speed implementations.
The Air Consumption AC is defined as the volume of air exploited in the cylinder at each cycle, function of source pressure $AC = pl(S + s)$, with l piston stroke.

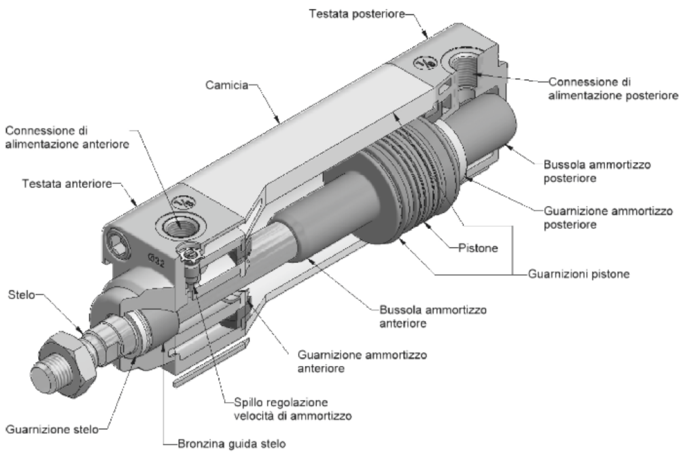


Figure 5.2: Double Effect Cylinder[11]

The maximum axial load $F_{a,max}$, is the maximum load applicable before a deflection of the piston's rod. This load is influenced by: Load, Rod diameter d, Distance of load application L, Cylinder mounting condition. The load can be calculated as

$$F_{a,max} = \frac{p^3 E d^4}{64 L^2 C} \tag{5.1}$$

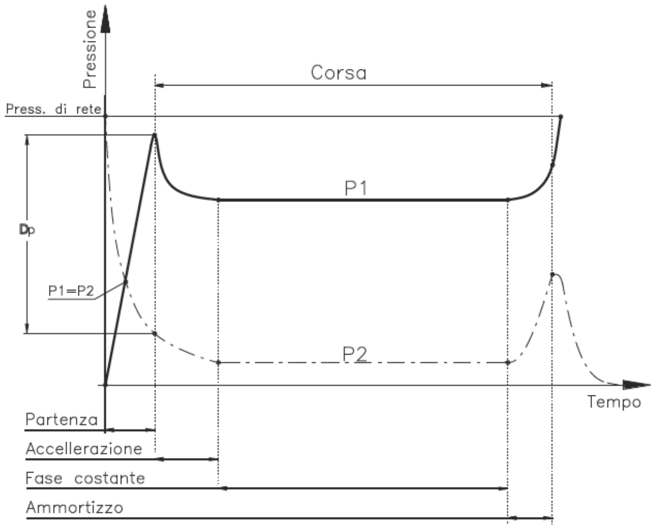


Figure 5.3: Double Effect Cylinder - Typical Graph[11]

With C safety factor and E Young elastic modulus.
To control air flow, a valve is requested. The valve distributes the compressed air and manages its flow, there are three main categories:

- Interception Valve: stop or vary the air flow.
- Regulation Valve: manages compressed air, adjusting pressure and flow rate.
- Distribution Valve: deviate compressed air, without influencing pressure or flow rate.

The valves can be classified by two methods of functioning:

- Spring Offset functioning
  The flow control is performed through rubber sealing. Advantages: short response time, reduced pressure drops, high flow rate allowed. Disadvantages: monostable functioning (return to neutral position through spring), pressure not balanced, 5/3 functioning not allowed.
- Plunger functioning:
  The flow control is performed through a plunger. Advantages: mounting simplicity, 5/3 functioning allowed, reduced dimensions, parallel layout allowed. Disadvantages: slower response time, slower mounting process, lower flow rate.

The valve required in solution 1 is a Bistable 5/3 functioning valve (5 ports, 3 positions), as the one reported in figure 5.4. The Neutral position is exploited during the preparation of test bench, the other two opposite positions, are exploited to generate force in one direction or the other. The drawbacks of the system just

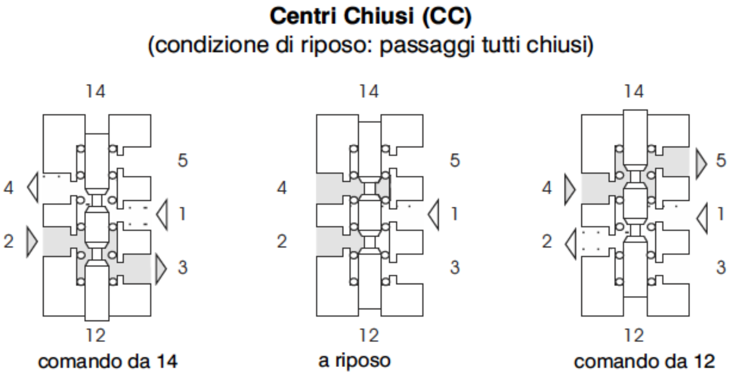


Figure 5.4: Plunger Valve 5/3 Functioning [11]

explained are here reported:

- Number of components: the pneumatic circuit to realize the system displayed, requires a number of components that affects the final cost.
- Friction losses: the seals of the pneumatic cylinder affect the force generated by the piston, with $10 - 15\%$ of losses.
- End stop: the end stop generates a peak force at stroke end. The peak force influences the characteristic of the gearshift system and limits the correct stroke of the cable-end.
- Reaction force adjustment: the control of reaction forces, exploiting compressed air, has not the precision needed. Valve speed, in controlling flow rate, is not sufficient to adjust the forces generated in real time with the required precision.

5.2 Reaction System - Solution 2

The concept of the second solution developed, is shown in figure 5.5. The idea is an improvement of the reaction system exploited today (see section 2.1.2).
The spring 12, compressed between plates 2 and 3 generates the reaction force, with the cable acting (axially respect to it) on sphere 9. During durability test, the gearshift system gets loose, efficiency changes, and to reach the same force target longer stroke is necessary. To maintain the correct stroke, producing the same force,

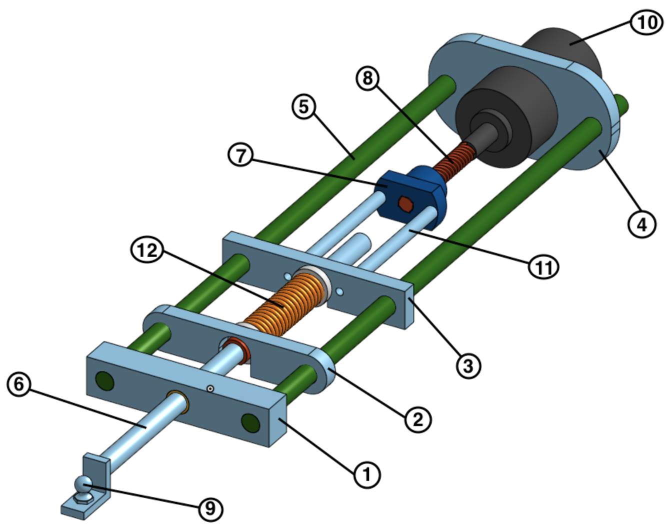


Figure 5.5: Reaction System - Solution 2

is required a variation of spring pre-load. To modify the pre-load a stepper motor 10 is exploited. To generate a linear motion from motor rotation, nut-screw system is adopted. The rotation of the screw 8, mounted on motor, generates the linear motion of the nut 7. The nut, through shafts 11, actuates the plate 3 and adjust the spring pre-load.

5.3 Solution 2 Design

The design of the reaction system go through the following steps:

- Dimensioning of Nut-Screw
- Dimensioning of Stepper motor
- Design of components

5.3.1 Dimensioning of Nut-Screw

The selection of nut-screw system is made on Igus catalogue[12], as reference manufacturer. Nut-screw system typically present a low efficiency, Igus to improve this aspect develops a high helix thread lead screw design. With this kind of thread, is expected higher efficiency due to optimized, flatter thread angle, and lower power loss is obtained in force transmission. The rounded teeth generate a reduced contact surface between nut and screw, which reduces the noise and vibrations transmitted. Each typology of nut is furnished in four different materials, depending on the application field. The material, selected for this project is *Iglidur J350*, material registered by Igus, which supports up to 150°C.

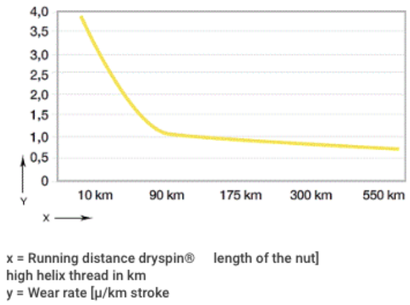
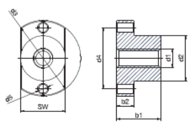


Figure 5.6: Nut Service Life - 175 N, 540 mm stroke, 125 rpm

Iglidur J350 is a polymer with high wear resistant and self-lubricating properties, which guarantees long service life, with a reduction of maintenance and lubrication cost.
In figure 5.7 is reported the list of allowable nut, with anti-rotation geometry and high helix thread. The axial load supported is the main parameter for the selection. Considering the explanation of working principle above, the component can be considered working statically, in supporting spring pre-load, with few small adjustment performed during test execution. The nut underlined is the one selected. Overdimensioned to guarantee the maintenance of plate 3 position, during alternated load applied by gearshift system, and to support spring charging load.



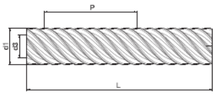
Technical data

Thread	Hand of rotation Right	Hand of rotation Left	Effective supporting surface [mm²]	Max. stat. axial F [N] iglidur® J / J350 / A180	Max. stat. axial F [N] iglidur® R
Ds6.35x2.54	●	–	199	125	125
Ds6.35x5.08	●	–	156	125	125
Ds6.35x12.7	●	–	78	125	125
Ds6.35x25.4	●	–	114	125	125
Ds8x10	●	●	203	507	406
Ds8x15	●	●	205	513	410
Ds10x12	●	●	343	858	686
Ds10x25	●	●	312	779	623
Ds10x50	●	●	253	632	506
Ds12x5	●	–	581	1,452	1,162
Ds12x25	●	–	562	1,405	1,124
Ds14x25	●	●	576	1,440	1,152
Ds14x30	●	–	571	1,427	1,142
Ds14x40.6	●	–	562	1,405	1,124
Ds16x35	●	–	668	1,669	1,335
Ds18x24	●	●	821	2,053	1,643
Ds18x40	●	●	764	1,911	1,529
Ds18x80	●	●	528	1,319	1,056
Ds18x100	●	●	463	1,158	926
Ds20x20	●	–	1,083	2,707	2,165
Ds20x60	●	●	729	1,822	1,458
Ds20x80	●	–	755	1,888	1,510
Ds20x90	●	●	671	1,678	1,342

Dimensions [mm]

Thread	d1	d2	d3	d4	d5	b1	b2	SW	Weight [g] iglidur® J	Weight [g] iglidur® J350	Weight [g] iglidur® R	Weight [g] iglidur® A180	Part No.
Ds6.35x2.54	6.35	13	25	19	3.2	15	5	13	3.7	3.6	3.4	3.6	DST-□FRM-131315DS6.35X2.54
Ds6.35x5.08	6.35	13	25	19	3.2	15	5	13	3.7	3.6	3.4	3.6	DST-□FRM-131315DS6.35X5.08
Ds6.35x12.7	6.35	13	25	19	3.2	15	5	13	3.7	3.6	3.4	3.6	DST-□FRM-131315DS6.35X12.7
Ds6.35x25.4	6.35	13	25	19	3.2	15	5	13	3.7	3.6	3.4	3.6	DST-□FRM-131315DS6.35X25.4
Ds8x10	8	20	36	28	4	20	8	20	12.7	12.3	11.8	12.4	DST-□F□M-202020DS8X10
Ds8x15	8	20	36	28	4	20	8	20	12.7	12.3	11.8	12.4	DST-□F□M-202020DS8X15
Ds10x12	10	25	42	34	5	25	10	25	23.7	22.9	22.1	23.2	DST-□F□M-252525DS10X12
Ds10x25	10	25	42	34	5	25	10	25	23.7	22.9	22.1	23.2	DST-□F□M-252525DS10X25
Ds10x50	10	25	42	34	5	25	10	25	23.7	22.9	22.1	23.2	DST-□F□M-252525DS10X50
Ds12x5	12	28	48	38	6	35	12	28	39.2	37.9	36.6	38.4	DST-□FRM-282835DS12X5
Ds12x25	12	28	48	38	6	35	12	28	39.2	37.9	36.6	38.4	DST-□FRM-282835DS12X25
Ds14x25	14	28	48	38	6	35	12	28	37.1	35.9	34.6	36.4	DST-□F□M-282835DS14X25
Ds14x30	14	28	48	38	6	35	12	28	37.1	35.9	34.6	36.4	DST-□FRM-282835DS14X30
Ds14x40.6	14	28	48	38	6	35	12	28	37.1	35.9	34.6	36.4	DST-□FRM-282835DS14X40.6
Ds16x35	16	28	48	38	6	35	12	28	34.6	33.5	32.3	33.9	DST-□FRM-282835DS16X35
Ds18x24	18	28	48	38	6	35	12	28	31.9	30.8	29.7	31.2	DST-□F□M-282835DS18X24
Ds18x40	18	28	48	38	6	35	12	28	31.9	30.8	29.7	31.2	DST-□F□M-282835DS18X40
Ds18x80	18	28	48	38	6	35	12	28	31.9	30.8	29.7	31.2	DST-□F□M-282835DS18X80
Ds18x100	18	28	48	38	6	35	12	28	31.9	30.8	29.7	31.2	DST-□F□M-282835DS18X100
Ds20x20	20	32	55	45	7	44	12	32	49.2	47.6	45.9	48.2	DST-□FRM-323244DS20X20
Ds20x60	20	32	55	45	7	44	12	32	49.2	47.6	45.9	48.2	DST-□F□M-323244DS20X60
Ds20x80	20	32	55	45	7	44	12	32	49.2	47.6	45.9	48.2	DST-□FRM-323244DS20X80
Ds20x90	20	32	55	45	7	44	12	32	49.2	47.6	45.9	48.2	DST-□F□M-323244DS20X90

Figure 5.7: Igus Nut table



Technical data

Thread	Hand of rotation Right	Hand of rotation Left	Material Stainless steel 1.4301	Material Aluminium EN AW 6082	Pitch P [mm]	Number of threads	Pitch angle α [°]	Weight Stainless steel [kg/m]	Weight Aluminium [kg/m]
Ds6.35x2.54	●	–	●	●	2.54	2	7.26	0.25	0.09
Ds6.35x5.08	●	–	●	●	5.08	4	14.29	0.25	0.09
Ds6.35x12.7	●	–	●	●	12.7	4	32.48	0.25	0.09
Ds6.35x25.4	●	–	●	●	25.4	8	51.85	0.25	0.09
Ds8x10	●	●	●	●	10	4	21.70	0.40	0.14
Ds8x15	●	●	●	●	15	6	30.83	0.40	0.14
Ds10x12	●	●	●	●	12	4	21.54	0.62	0.21
Ds10x25	●	●	●	●	25	8	38.51	0.62	0.21
Ds10x50	●	●	●	●	50	10	57.86	0.62	0.21
Ds12x5	●	–	●	●	5	2	7.55	0.89	0.31
Ds12x25	●	–	●	●	25	8	33.55	0.89	0.31
Ds14x25	●	●	●	●	25	5	29.61	1.22	0.42
Ds14x30	●	–	●	●	30	6	34.30	1.22	0.42
Ds14x40.6	●	–	●	●	40.6	8	42.71	1.22	0.42
Ds16x35	●	–	●	●	35	7	34.85	1.59	0.54
Ds18x24	●	●	●	●	24	6	22.99	2.01	0.69
Ds18x40	●	●	●	●	40	8	35.55	2.01	0.69
Ds18x80	●	●	●	●	80	12	54.74	2.01	0.69
Ds18x100	●	●	●	●	100	12	60.51	2.01	0.69
Ds20x20	●	–	●	●	20	4	17.66	2.48	0.85
Ds20x60	●	●	●	●	60	8	43.68	2.48	0.85
Ds20x80	●	–	●	●	80	12	55.07	2.48	0.85
Dx20x90	●	●	●	●	90	12	55.08	2.48	0.85

Dimensions [mm]

Outer Ø d1 -0.05	Core Ø d3 -0.05	Max. total length L Stainless steel	Max. total length L Aluminium	Part No.
6.35 -0.1	4.35 -0.1	1,000	1,000	DST-LS-6.35X2.54-R-▭-ES
6.35 -0.1	4.85 -0.1	1,000	1,000	DST-LS-6.35X5.08-R-▭-ES
6.35 -0.1	4.35 -0.1	1,000	1,000	DST-LS-6.35X12.7-R-▭-ES
6.35 -0.1	4.15 -0.1	1,000	1,000	DST-LS-6.35X25.4-R-▭-ES
8.0	5.63	1,000	1,000	DST-LS-8X10-□-▭-ES
8.0	5.63	1,000	1,000	DST-LS-8X15-□-▭-ES
10.0	6.95	3,000	1,000	DST-LS-10X12-□-▭-ES
10.0	7.10	3,000	1,000	DST-LS-10X25-□-▭-ES
10.0	7.35	3,000	1,000	DST-LS-10X50-□-▭-ES
12.0	9.60	3,000	1,500	DST-LS-12X5-R-▭-ES
12.0	8.95	3,000	1,500	DST-LS-12X25-R-▭-ES
14.0	9.60	3,000	1,500	DST-LS-14X25-□-▭-ES
14.0	9.60	3,000	1,500	DST-LS-14X30-R-▭-ES
14.0	9.65	3,000	1,500	DST-LS-14X40.6-R-▭-ES
16.0	11.60	3,000	1,500	DST-LS-16X35-R-▭-ES
18.0	14.40	3,000	1,500	DST-LS-18X24-□-▭-ES
18.0	13.60	3,000	1,500	DST-LS-18X40-□-▭-ES
18.0	14.00	3,000	1,500	DST-LS-18X80-□-▭-ES
18.0	13.55	3,000	1,500	DST-LS-18X100-□-▭-ES
20.0	15.60	3,000	1,500	DST-LS-20X20-R-▭-ES
20.0	15.55	3,000	1,500	DST-LS-20X60-□-▭-ES
20.0	15.98	3,000	1,500	DST-LS-20X80-R-▭-ES
20.0	15.55	3,000	1,500	DST-LS-20X90-□-▭-ES

Figure 5.8: Igus Screw table

The screw, with high helix thread, coupled with it, is available in Aluminium or Stainless Steel. The stainless steel screw is selected, in fig. 5.8, to guarantee higher reliability of the component, and improve resistance of the system to oxidation.

5.3.2 Dimensioning of Stepper motor

For the selection of the stepper motor, first is necessary define the torque and the power required by the system to work properly. Here are reported the equations to calculate these parameters, and the data linked to the reaction system designed.

$$M = \frac{F_a p}{2\pi\ \eta 1000} \tag{5.2}$$

$$P = \frac{Mn}{9.55} \tag{5.3}$$

The data needed to perform the calculations:

- Axial Load F_a: When the spring adopted is dimensioned for the specific gearshift system, an initial pre-load is not required. Depending on project specification, the reaction force required may vary. To maintain a safety threshold in motor dimensioning, is chosen a value of axial load $F_a = 350\ N$. This value takes into account even the force required to compress the spring to adjust the pre-load.

- Pitch p: The value of pitch is obtained from tables on Igus catalogue [12], $p = 12\ mm$.

- Efficiency η: Nut-Screw system typically presents very low values of efficiency, $\eta = 0.2 \div 0.4$, due to high friction behind the working principle of the system.

The high helix thread design, improves this parameter, the nut-screw system selected presents, in dry conditions, efficiency $\eta = 0.5 \div 0.8$. The value selected to perform calculations is $\eta = 0.65$.

- Rotational Speed n: The system designed, doesn't requires a continuous rotation of the screw. As consequence the wear of components is very low, and the influence of rotational speed is minimum. One turn of the screw ($360deg$), defines a linear displacement of the nut equal to the pitch ($12\ mm$). When the adjustment of spring pre-load is required, the single shift of plate 3 is limited to a value much lower of the pitch. The rate of the displacement isn't a fix value, it depends on the system tested. To select an oversized motor, is considered a rotational speed $n = 3\ rps = 36\ mm/s \approx 0.04\ m/s$.

Form equations 5.2 and 5.3, are required a torque $M = 1.028\ Nm$ and a power $P = 0.004\ kW$.
Igus provides a series of Electric motors, studied to interface Nut-Screw system. The selection of the motor is performed on Igus Catalogue for Lead Screw Stepper Motor [14]. Stepper Motor Nema23, selected in figure 5.10, results the one that fits the required performances (Torque, Axial Load supported, Angular step).

Technical data							
Flange dimension		**28**	**28**	**42**	**42**	**56**	**56**
Specification		**S**	**M**	**S**	**M**	**S**	**M**
Max. voltage	[VDC]	60	60	60	60	60	60
Nominal voltage	[VDC]	24-48	24-48	24-48	24-48	24-48	24-48
Nominal current	[A]	0,67	1,0	1,4	1,8	2,8	4,2
Holding torque	[Nm]	0,06	0,12	0,2	0,5	1,0	2,0
Detent torque	[Nm]	0,0025	0,004	0,006	0,022	0,03	0,068
Step angle	[°]	1,8±5%	1,8±5%	1,8±5%	1,8±5%	1,8±5%	1,8±5%
Resistance/Phase	[Ω]	5,60±15%	2,30±15%	2,0±15%	1,75±15%	0,75±15%	0,50±10%
Inductivity/Phase	[mH]	4,00±20%	1,80±20%	3,0±20%	3,30±20%	2,60±20%	2,20±20%
Rotor inertia	[kgcm²]	0,009	0,018	0,036	0,082	0,27	0,48
Shaft load, axial	[N]	50	50	100	100	500	500
Shaft load, radial	[N]	-	-	-	-	-	-

Figure 5.9: Stepper Motor Igus Selection [14]

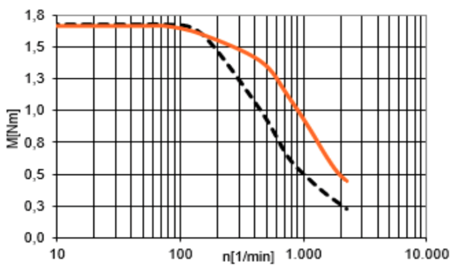
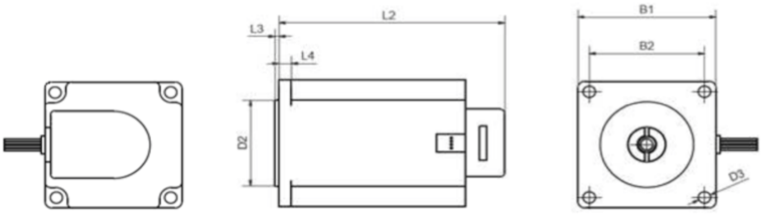


Figure 5.10: Stepper Motor Igus Nema23 Characteristic [14]

Moreover the limited dimensions of the motor, and the value of *step angle*, represent two additional advantages that make it the ideal solution for the designed system.



With Encoder	Specification	B1 [mm]	B2 [mm] ±0,2	D2 Ø [mm] ±0,025	D3 Ø [mm]	D4 Ø [mm]	D5 Ø [mm]	L2 [mm] ±1	L3 [mm]	L4 [mm]
MOT-ST-28-L-C-B	S	28,2	23,00	22,00	M2,5	-	-	47,2	2,0	0
MOT-ST-28-L-C-A	M	28,2	23,00	22,00	M2,5	-	-	66,2	2,0	0
MOT-ST-42-L-C-B	S	42,3	31,00	22,00	M3	-	-	46,2	2,0	0
MOT-ST-42-L-C-A	M	42,3	31,00	22,00	M3	-	-	65	2,0	0
MOT-ST-56-L-C-B	S	56,4	47,14	38,10	5	-	-	65,7	1,6	5
MOT-ST-56-L-C-A	M	56,4	47,14	38,10	5	-	-	92	1,6	5

Figure 5.11: Stepper Motor Igus Nema23 Dimensions [14]

5.3.3 Design of components

This step is related to the interface between components, and technical solution necessary to assembly the system. The design, performed on CAD software, allows to check dimensions and mounting issue. The main features to consider, during this step, are:

- Guarantee the necessary space for components motion.
- Guarantee reliable and easy assembly.
- Guarantee flexibility of the system (adaptation to different projects).
- Avoid manufacturing of useless components.

Here is reported the schematic functioning of the reaction system just exposed. With blue arrows are indicated the feedback signals, useful to generate a voltage signal from workstation, and adjust stepper motor position. The workstation checks Force-Stroke parameters at the actuation of gearshift system, and corrects spring pre-load to maintain nominal Force-Stroke at cables output.

V_1 Voltage input to e.m.
V_2 Voltage input to Stepper motor
F_{em} Force input to Actuation system
s_{em} Stroke input to Actuation system
F_{act} Force input to Gearshift system
s_{act} Stroke input to Gearshift system
F_{grs} Force output from Gearshift system
s_{grs} Stroke output from Gearshift system
s_{sm} Stroke input to Nut-Screw
F_{ns} Force input to spring (reaction)
s_{ns} Stroke input to spring (reaction)
F_{reac} Force Feedback from reaction
F_{act1} Force Feedback from Actuation system
s_{act1} Stroke Feedback from Actuation system

Figure 5.12: Reaction System - Solution 2 Schematic Functioning Parameters

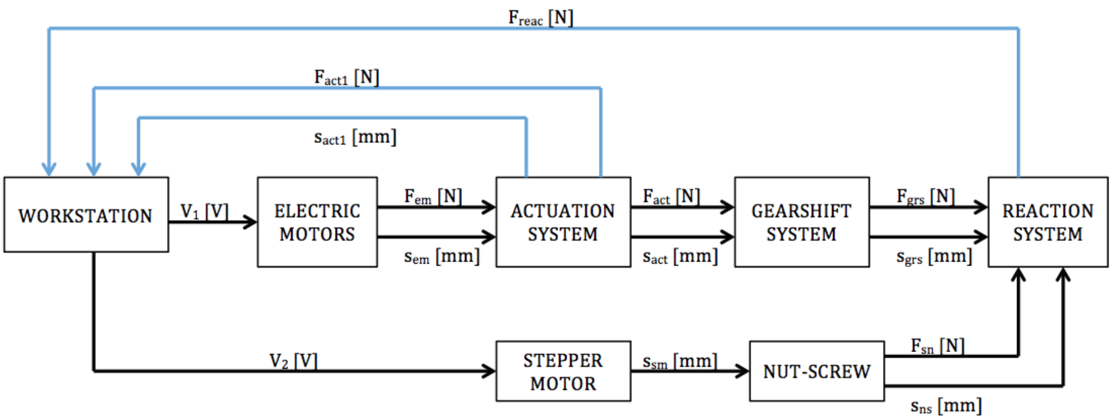


Figure 5.13: Reaction System - Solution 2 Schematic Functioning

Chapter 6

Conslusions and Further Developments

The objective of this work, was the design of a new Actuation system and Reaction system for test bench, in testing department of Sila Holding Industriale SpA. The new solutions developed should go over limits of existing systems, and respecting defined performances, improve the reliability and robustness of test benches.
The work was focused firstly on the Actuation system, with the design of three solutions, and the identification of their advantaged and drawbacks. A fourth solution, resulted the best trade off to obtain the improvement desired in performance and reliability of the system. However this solution was designed with the purpose to limit as much as possible the costs, and to obtain rapidly a prototype. As consequence some components, and technical solutions adopted, were selected between parts already available in the company.
A further development for the Actuation system, regards the modification of the sliding system in Select direction. In particular shafts 9 and related parts will be substituted by a Linear Guide (as the one adopted for rack), properly dimensioned to sustain load acting on the system. The linear guide will guarantee: longer service life, lower frictional losses, higher packaging, higher reliability.
The second point, was the design of the Reaction system. The first solution proposed had drawbacks, mainly concerning cost and performances, solved with the improvement of existing system in solution 2. The second solution, allowed to reach the objective with minor modifications of actual reaction system. A further step is required on this point, not developed in this work, because argument of study of a different department. It is necessary to interface the functioning of the modified reaction system, with the software exploited to control test benches and perform tests, in accordance with scheme illustrated in fig. 5.13.
The final step for this work of thesis was the Realization and Validation of the solutions designed. Work commitments of the company and availability of test benches

are slowing down the assembly process of the prototypes for the validation. The first components, figure 6.1, have been manufactured and validation test will be carried out on Suzuki gearshift system.

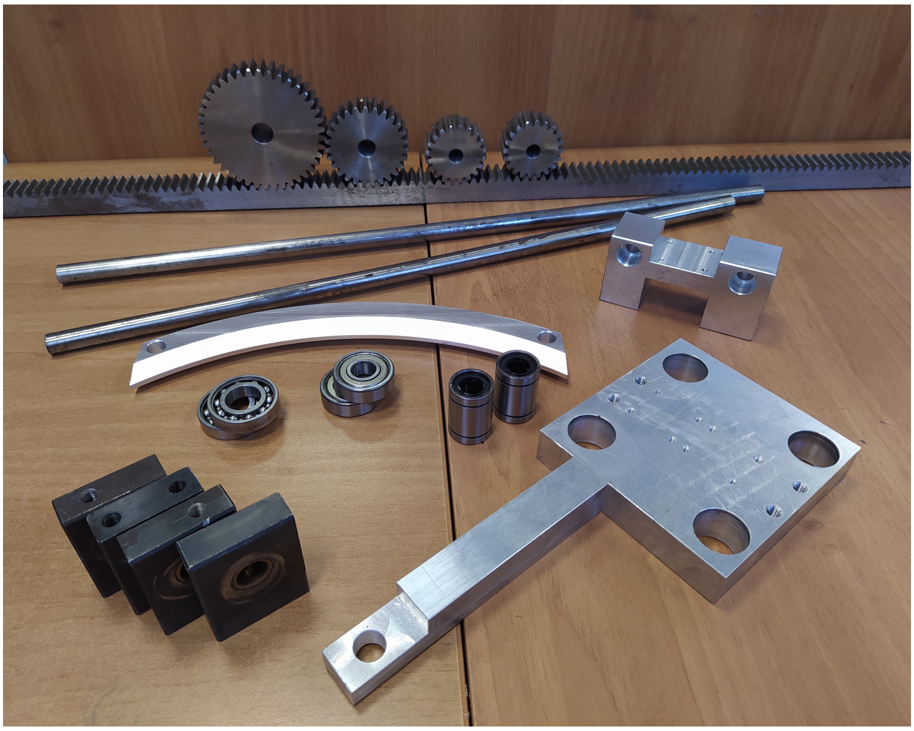


Figure 6.1: Actuation System Solution 4 - Manufactured Components

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