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Study and Design of a Test Bench for Balancing and Diagnosis of Faults

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Abstract: *During manufacturing, imperfections significantly impact the functioning of rotating machinery. Often the centers of rotation and of mass will not coincide perfectly, causing a mass imbalance in the machine. Imbalance is the most common problem in the rotating machinery industry. It causes vibrations, breakdowns in machines, untimely stoppages, and disturbances in production in general. The search for solutions to minimize the level of machine vibration is more than necessary. The objective is to design and manufacture special machines to locate imbalances, and other defects.*
The article is devoted to the study, the design and the realization of a test bench for balancing and diagnosis of faults. The latter helps to take measurements and analyze vibrations to find better solutions. For example, engines, motors, discs, and turbines develop characteristic inertia effects that can be analyzed to improve the design and decrease the possibility of failure.
We perform both theoretical and experimental studies to achieve a sustainable realization which helps to take measurements and analyze vibrations to find better solutions. In this paper, the processes of mechanical design, 3D designs, and the realization of the test bench are tackled.

Keywords: *Imbalance, Vibration, reducing vibration, rotating machines, test bench, technological system.*

1. Introduction

Vibration analysis allows evaluating the health of equipment. The goal is to minimize unplanned downtime by finding failures. Vibration can be a sign and a source of trouble. Vibration can result from a number of conditions, acting alone or in combination: Imbalance, Misalignment, Wear, and Looseness.

The aim of this paper is the study of a balancing test bench. This wheel balancer is modular with interchangeable parts.

Unbalance is the most common problem in the rotating machinery industry. It causes vibrations, breakdowns in machines, untimely stoppages, and disturbances in production in general. We propose to study a balancing test bench in this work.

Balancing allows reducing vibration to improve performance and reliability of the machines..

There are several types of Unbalances [1]: Static Unbalance, couple Unbalance, Quasi-Static Unbalance, and Dynamic unbalance.

2. Mechanical Design

Machine design is defined as the use of scientific principles, technical information and imagination in the description of a machine or a mechanical system to perform specific functions with maximum economy and efficiency. This definition of machine design contains some important features.
The final outcome of the design process consists of the description of the machine **[2].**
the kinematic chain is given in Fig. 1.

Kinematic chain

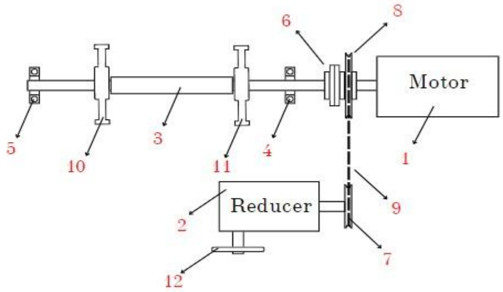


Fig. 1: kinematic chain

1. Motor: P = 1,5 KW N = 2895 tr /min
2. Reducer: Ratio = 1: 80
3. Shaft: L = 600 mm $Ø_1$ = 40 mm $Ø_2$ = 30 mm
4. & 5. Bearings Block: Ref: ISO 16949 UCP206
6. Coupling
7. Driven pully: Ø = 160 mm
8. Driving Pully: Ø = 80 mm
9. Belt: Type: V-belt L= 1150mm Ref: 12.5X1150 La
10. & 11. Disc: Ø = 300 mm
12. Roller Chain Sprocket

2.1. Shaft design: (Fig.2)

The term ‘transmission shaft’ usually refers to a rotating machine element, circular in cross section, which supports transmission elements like gears, pulleys, discs and transmits power.

The shaft is always stepped with maximum diameter in the middle portion and minimum diameter at the two ends, where bearings are mounted. The steps on the shaft provide shoulders for positioning the two discs and bearings **[2], [3].**

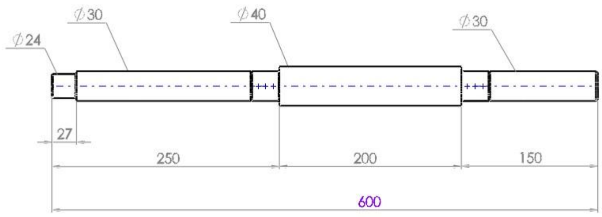
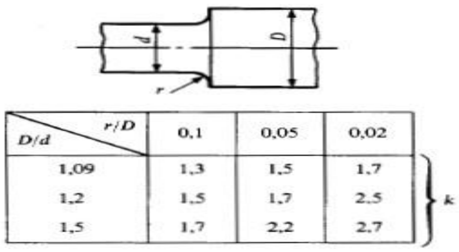


Fig.2: Shaft dimensions

By using the notions of Resistance of materials and after mathematical considerations, we find Torsional shear stress calculation and Stress concentration TABLE I and Torsional Verification and Minimum shaft diameter TABLE II.

TABLE I: Torsional shear stress calculation and Stress concentration

Applied torque calculation , Torsional shear stress calculation and Stress concentration, Torsional shear stress calculation and Stress concentration		
Application	Given or data	Symbols
Mt = 4947.28 N. mm d = 30mm → τ=0,933 N/mm2 d = 40mm → τ=0,324 N/mm2 d = 25mm → τ=1,612 N/mm2 We take the biggest result for safety	P = 1,5 KN N=2895 tr/min $r = \frac{d}{2}$ $Io = \frac{\pi d^4}{32}$ $Io/_r = \frac{\pi d^3}{16}$	τ : torsional shear stress (N/mm2) Mt : applied torque (N.mm), d : shaft diameter (mm) Re : Yield Strength (N/mm2) Rpg : practical shear strength (N/mm2) S : Factor of safety , P : power (KN), r : Radius N : Speed of rotation (tr/min), Io : moment of inertia r : Radius K : Stress concentration
$\frac{r}{d}$ = 0,02 $\frac{D}{d}$ = 1,33 K = 2,7	See figure 11 **[3]**	r = 0,8 D = 40 d = 30



D/d \ r/D	0,1	0,05	0,02
1,09	1,3	1,5	1,7
1,2	1,5	1,7	2,5
1,5	1,7	2,2	2,7

Fig.3: Stress concentration [3]

TABLE II: Torsional Verification and Minimum shaft diameter

Formulas	Application	Given or data
$\text{Rpg} = \frac{\text{Re}}{\text{S}}$	$\text{Rpg} = 177{,}5\ \text{N/mm2}$	τ_e : Elastic shear stress (N/mm2) τ_p : practical shear stress (N/mm2) $\tau = 1{,}612\ \text{N/mm2}$ $\text{Re} = 355\ \text{N/mm2}$ (S355) S = 2 k = 2,7 $\tau . \text{k} \leq \text{Rpg}$ $\tau . \text{k} \leq \tau_p$ The condition is verified.
$\tau\text{max} =$ $\tau . \text{k} \leq \text{Rpg}$	$\tau\ \text{max} = 4{,}353\ \text{N/mm2}$ $4{,}353 \leq 177{,}5$	
$\tau_e = 0{,}7 . Re$ $\tau_p = {}^{\tau_e}/_2$	$\tau_e = 248{,}5\ \text{N/mm2}$ $\tau_p = 124{,}25\ \text{N/mm2}$	
$d = \sqrt[3]{\frac{16 . Mt}{\pi . \tau\ \text{max}}}$	d = 17,95 mm	$\tau\ \text{max} = 4{,}353\ \text{N/mm2}$ $\text{Mt} = 4947.28\ \text{N.mm}$

2.2. Bending strength verification "Flexional resistance"

We give the shaft loads schema (Fig.4) from which we calculate Mass calculation and Loads calculations (Table III) and Bending moment & shear force (TABLE IV) . Also, we give Bending moment & shear force diagram (Fig.5).

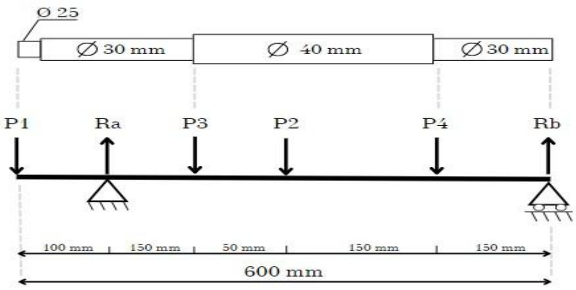


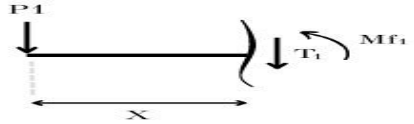
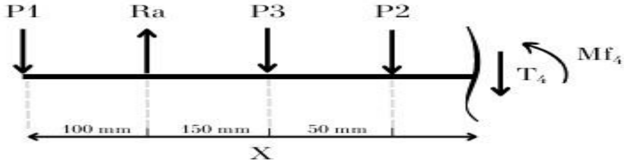
Fig.4: shaft loads schema

Table III: Mass calculation and Loads calculations

$\rho = \frac{m}{V}$ ρ = 7800 Kg/m3	ρ : Volumic mass (Kg/m3) m : mass (kg) V : Volume (mm3) S : Area (mm2) l : length (mm)
$V = S \times l$ = 528239,75 mm3	
$m = \rho . V$ =4,16 Kg	
P1 = 4,90 N P2 = 40,81 N P3 = 72,88 N P4 = 72,88N Ra= 103,246 N Rb=88,223	P1 : Coupling load (N) P2 :Shaft load (N) P3 : Disc1 load (N) P4 : Disc2 load (N) Ra : Reaction on point A (N) Rb : Reaction on point B (N) g : Gravity (N/Kg) m1 : Coupling mass (Kg) m2 : Shaft mass (Kg) m3 : Disc1 mass (Kg) m4 : Disc2 mass (Kg) m1 = 0,2 Kg m2 = 4,16 Kg m3 = m4 = 0,45 Kg

We consider the different sections for which we calculate the Bending moment Mf and the shear force T (TABLE IV). (Example TABLE 10 Bending moment & shear force)
Section 1 → 0≤X ≤100 , Section 2 →100≤X ≤250, Section 3 →250≤X ≤300,
Section 4 300≤X ≤450, Section 5, 450≤X ≤600

TABLE IV: Bending moment & shear force

Considered section	Plan & application	Application
Section 1 $0 \le X \le 100$	 $\sum \vec{F}\,ext = 0$ $\quad T_1 = -4{,}90\,N$ $\sum \vec{M}/_X = \vec{0}$ $\begin{cases} if\ X = 0\,; Mf_1 = 0\ \ N.mm \\ if\ \ X = 100\,;\ Mf_1 = -490\,N.mm \end{cases}$	P1 = 4,90 N P2 = 40,81 N P3 = 72,88N P4 = 72,88 N Ra = 103,246 N Rb = 88,223 N Mf : Bending moment T : shear force
Section 4 $300 \le X \le 450$	 $\sum \vec{F}\,ext = 0$ $\quad T_4 = -15{,}344\,N$ $\sum \vec{M}/_X = \vec{0} \begin{cases} if\ X = 300\,; Mf_4 = 15535{,}2\,N.mm \\ if\ X = 450\,;\ Mf_4 = 13233{,}6\,N.mm \end{cases}$	

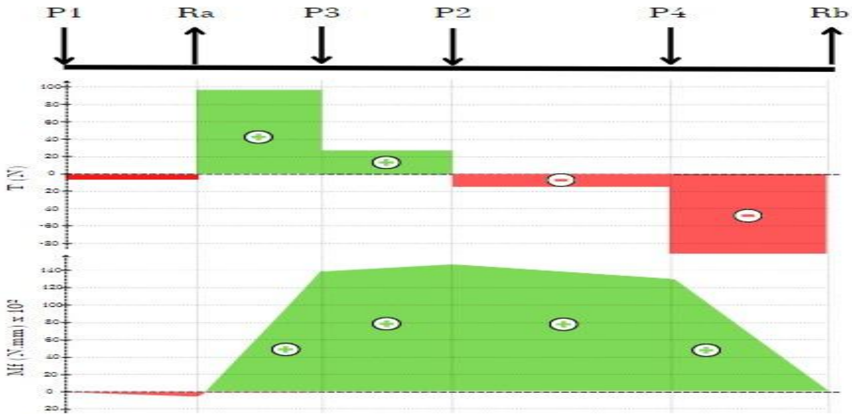


Fig.5: Bending moment & shear force diagram

In the same way we calculate Flexional resistance (*TABLE V)*

TABLE V: Flexional resistance

	$w\ (mm^3)$	σ (N/mm^2)	σ . K	Condition verification	Given or data
0~100	2650,718	−0,18	−0,486	Checked	σ : bending stress (N/mm2) σmax : Maximal bending stress (N/mm2) Mf : Bending moment (N.mm) d : shaft diameter (mm) Re : Yield Strength (N/mm2) Rpg : practical shear strength (N/mm2) S : Factor of safety K : Stress concentration, K= 2,7 W : bending modulus, Rpg = 177,5 N/mm^2
100~250	2650,718	5,380	14,526	Checked	
250~300	6283,185	2,472	6,674	Checked	
300~450	6283,185	2,472	6,674	Checked	
450~600	2650,718	4,992	13,479	Checked	

Note: we neglected the diameter with the coupling will be in (Ø 25)

3. Motor's Basics *(Fig.6)*

The purpose of a motor, regardless of the application of the project, is to change electrical power to mechanical power in order to provide rotational movement. Every application will have its own distinct parameters for input and output power. The Figure 6 provides a visual representation of the input and output parameters of a motor. The output power is the motor speed and torque response required to accomplish the task **[7].**

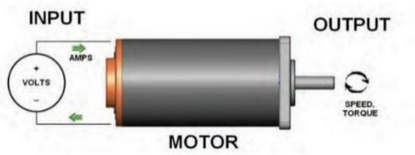


Fig.6: Motor Input and Output Functions **[3]**

3.1 Motor Selection:

The motor selection process begins with evaluating the application and ensuring the chosen motor will adequately match the application's needs. Though often overlooked by design engineers, the items on the Application Considerations Checklist (Figure7) are critical to motor design **[4].**

Fig.7: Application /Considérations Check-list **[10]**

- **Input Power Source:** The input power will be a known quantity and is easy to specify in the form of voltage, current, and frequency.
- **Environment:** Ambient temperature is also an important factor to take into consideration when choosing a motor

  To more uniformly denote industry-standard Gear motor sealing, the Ingress Protection (IP) chart (figure8) assists designers with selecting the proper IP rating.

INGRESS PROTECTION [IP] RATINGS			
FIRST NO.	PROTECTION AGAINST SOLIDS	SECOND NO.	PROTECTION AGAINST LIQUIDS
0	No protection	0	No protection
1	Objects over 50 mm (hand)	1	Vertically falling drops of water
2	Objects over 12 mm (finger)	2	Direct sprays up to 15° from vertical
3	Objects over 2.5 mm (tools and wires)	3	Direct sprays up to 60° from vertical
4	Objects over 1 mm (small tools and wires)	4	Sprays from all directions, limited ingress
5	Dust-limited ingress (no harmful deposit)	5	Low pressure jets of water from all directions, limited ingress
6	Totally protected against dust	6	Strong jets of water from all directions, limited ingress
		7	Temporary immersion (30 minutes) between 15 centimeters and 1 meter
		8	Long periods of immersion under pressure

Fig.8: Ingress Protection (IP) Rating Chart **[10]**

- **Motor Specs:** when considering new projects, one of the first considerations should be the design of the motor housing. In order to meet the needs of the application, the designer must understand the size and weight restrictions of the product.

- **Motor Performance:** Motor performance has been broken down into three key parameters:
- Speed and Torque, Starting/Stall Torque and Duty Cycle or Load Profile

Speed and torque represent the output that will be required to power the application and will affect the size of the motor. The motor can be optimized in the application by devoting special attention to the rating and operating characteristics of the motor design.

For our study, we choose an alternating current motor *(Fig.9)*.

Fig.9: AC Motor Constructions **[10]**

Characteristics:

- Relatively fixed speed operation (without a control), typically limited to 3,400 rpm
- Efficiency 60%-90% (small to large)
- Low (single phase) to medium (three phase) starting torque
- Operates on AC line voltage
- Life 20,000+ hours
- Totally enclosed construction is typical

The AC Induction Motor, depending on the construction, can run off of either a single phase or a three-phase AC power source. The output speed of the motor is fairly fixed and is dependent only on the number of poles with which the stator is wound and the frequency of the input voltage. The value of the input voltage does not affect the speed, only the frequency of the input source.

4. Reducer Kinematic Chain (Fig.10)

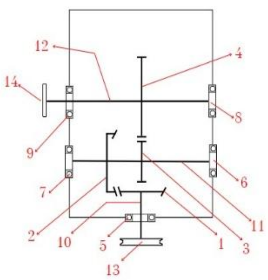


Fig.10: Reducer kinematic chain

1. Attac gear with a bellcranck :
   $Z_1 = 6$
   $\varnothing_1 = 16$ mm
2. Cog-wheel with bellcrank :
   $Z_2 = 59$
   $\varnothing_2 = 122$ mm
3. Attac gear (Helical gear) :
   $Z_3 = 11$
   $\varnothing_3 = 24$ mm
4. Cog-wheel :
   $Z_4 = 87$
   $\varnothing_4 = 163$ mm
5. Bearing :
   Ref : not accessible
6. & 7. Bearings :
   Ref : 7304 = 580n20
8. & 9. Bearings :
   Ref : 6208 A – FAG10.Shaft :
   $L_{10} = 120$ mm
   $\varnothing_{10} = 19$ mm
11. Shaft :
   $L_{11} = 90$ mm
   $\varnothing_{11} = 19$ mm
12. Shaft :
   $L_{12} = 190$ mm
   $\varnothing_{12} = 41{,}3$ mm
13. Pulley :
   $\varnothing_{13} = 160$ mm

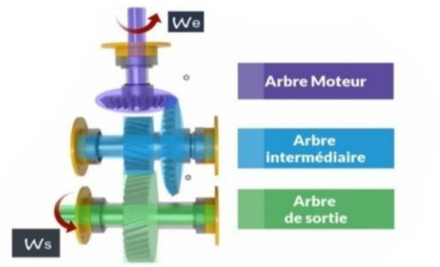
We give some pictures of our test bench and its components *photos 1to 3*

Photo 2 reducer case Photo2 reducer upper view

Photo 3Reducer's Parts

4.1. Ratio calculation

The calculations are based on the diagram in figure



Arbre moteur= Motor shaft

Arbre intermédiaire = Intermediate shaft

Arbre de sortie = Output shaft

Fig.11: Reducer schema

As we know that, $r = \frac{\omega_{input}}{\omega_{output}}$ $$r = (-1)^k \frac{\prod Z\ driving\ gear}{\prod Z\ driven\ gear} \quad (1)$$

$$r = (-1)^k \frac{Z1+Z3}{Z2+Z4} \qquad r = (-1)^2 \frac{6+11}{59+87}{}_{2} \qquad r = 0,12$$

K: number of exterior contacts Z: number of gear's teeth

4.2. V-Belt, Pulleys Sizing

V belts (also style V-belts, vee belts, or, less commonly, wedge rope) solved the slippage and alignment problem.

a) Factors that lead to belt failure:

- Poorly or Improperly Maintained:
- Design Flaws:
- Incorrectly Installed:
- The Environment:
- 5Defects and Poor Handling:

https://doi.org/10.17758/HEAIG11.H1122637

We wanted to reduce the output speed motor to the half of it so the ratio will be 1:2, by that we calculated the diameter of our pulleys, and the chosen diameter for the driving pulley is 80mm

$$\frac{\mathrm{d}}{\mathrm{D}} = r = \frac{1}{2} \qquad D = d2 = 2 \times 80 = 160mm \qquad (2)$$

D: driven pulley (mm) r: Ratio d: driving pulley (mm)

b) Selection of V-Belt "Sizing": In practice, the designer has to select a V-belt from the catalogue of the manufacturer. The following information is required for the selection [2]:

- Type of driving unit, Type of driven machine, Operational hours per day, Power to be transmitted, Input and output speeds, Approximate center distance depending upon the availability of space

The basic procedure for the selection of V-belts consists of the following steps:

- Determine the correction factor according to service (Fa) from the Table. It depends upon the type of driving unit, the type of driven machine and the operational hours per day.
  Based on the table (standards) **[9]** $F_a = 1{,}0$
- Calculate the design power by the following relationship:
  $P_D = P \times F_a$ (3)
  $P_D = 1{,}5\ (KW) \times 1{,}0 \qquad P_D = 1{,}5\ KW$
- Plot a point with design power as X coordinate and input speed as *Y* co-ordinate (Selection of Cross-section of V belt [9]). The location of this point decides the type of cross-section of the belt. Our belt will be in the A cross-section
- Determine the recommended pitch diameter of the smaller pulley from [2]. It depends upon the cross-section of the belt.
  Based in the Data we can select the following: Pitch Width W_P (mm) = 11 mm ,Nominal Top Width W (mm) = 13 mm, Nominal Height T (mm) = 8 mm
  Recommended Minimum Pitch diameter of pulley (mm) = 125 mm
- Calculate the Velocity of our system and output speed N2:

$$V = \frac{\pi\, D_1\, N_1}{60}$$

V: Velocity m/s D: Diameter of the driving pulley (mm) N: Speed of rotation (tr/min)

$$V = \frac{\pi \,.\, 80 \,.\, 2895}{60} \quad V = 12126{,}547\ mm/\ min \quad V = 0{,}727\ m/s$$

$$N_2 = \frac{N_1 \,.\, D_1}{D_2} = \frac{2895 \,.\, 80}{160} = 1447{,}5\ tr/min$$

- Determine the pitch length of belt *L* by the following relationship:

$$L_P = 2C + \frac{\pi}{2}\,(D_1 + D_2) + \frac{(D_1 - D_2)^2}{4C} \qquad (5)$$

L_P: pitch length of belt (mm) C: Center diameter between two pulleys (mm)
D_1: Bigger pulley diameter (mm) D_2: Smaller pulley diameter (mm)

$$\mathrm{L_P} = 2.390 + \frac{\pi}{2}\,(80 + 160) + \frac{(80 - 160)^2}{4.390} \qquad \mathrm{L_P} = 1161{,}09\ \mathrm{mm}$$

Now we need to select the standard pitch length (Nominal pitch lengths for standard sizes of V-belts **[2]).**

$$\mathrm{L_P} = 1100\ \mathrm{mm}$$

But after looking in our market we found out that there is: $\mathrm{L_P} = 1150\ \mathrm{mm}$

- Find out the correct center distance *C* by substituting the above value of *L* in the following equation:

$$L_P = 2C + \frac{\pi}{2}\,(D_1 + D_2) + \frac{(D_1 - D_2)^2}{4C} \qquad (6)$$

$$C^2 - 386{,}505\, C + 800 = 0 \qquad C = 384\ mm$$

- Correction factor (Fc) for belt pitch length depends upon the type of cross-section and the pitch length of the belt. Correction factors for belt pitch length (Fc**) [2].**
  we have $F_C = 0{,}90$
- Calculate the arc of contact for the smaller pulley by the following relationship:

$$\theta_S = 180 - 2\sin^{-1}\left(\frac{D_1 - D_2}{2C}\right) \quad \theta_S = 180 - 2\sin^{-1}\left(\frac{160-80}{2.384}\right) \tag{7}$$

$$\theta_S = 168{,}04°$$

- Determine the correction factor (Fd) for the arc of contact .
  We can see that our angle is near to 169°, so we can take $F_d = 0{,}97$

- Depending upon the type of cross-section, we determine the power rating (Pr) of single V-belt. It depends upon three factors—speed of faster shaft, pitch diameter of smaller pulley and the speed ratio.(Power ratings in kW (Pr) for A-Section V- Belts [2])

We have: $P_r = 2{,}55\ KW$

- The last step in the selection procedure is to find out the number of belts. It depends upon the design power and the power transmitting capacity of one belt. The number of belts is obtained by the following relationship:

$$N = \frac{P \times F_a}{P_r \times F_C \times F_d} \quad N = \frac{1{,}5 \times 1}{2{,}55 \times 0{,}90 \times 0{,}97} \quad N = 0{,}67 \quad N \approx 1 \tag{8}$$

By that only one belt will be enough for our operation

- Passage frequency: **[5]**

$$F_r = \frac{V}{L_p} = \frac{727}{1150} = 0{,}632\ S^{-1} \tag{9}$$

V: Velocity mm/s L_P: pitch length of belt (mm)

c) Coupling: The shafts to be connected by the coupling may have collinear axes, intersecting axes, or parallel axes with a small distance in between.

And in our case, we will be using a flexible coupling.

To make an informed choice, it is important to know the capabilities and differences between the different types of coupling. We choose the Jaw coupling.

Jaw coupling is a material flexing coupling. It finds use in general low-power transmission and motion control applications. It can accommodate any angular misalignment.

Reasons to Consider Using Jaw Couplings: Balanced, Zero-Backlash Design, Elastomer Spiders, High Customizability, Hub Separation, Fail Safe Design.

Selecting a Jaw Coupling model:

Step1: Determine the Nominal Torque of your application: [6]

Step 2: select the service factor which best corresponds to our application [6]

Step 3: Calculate the Design Torque

Step 4: Using the Spider Performance Data, we select the elastomer material which best corresponds to our application (Spider Performance Data).

Step 5: Using the Jaw Nominal Rated Torque we can choose our coupling size in the first column.

By following these steps, we find that the appropriate model for our project is L/AL090 [**6].**

d) Bearings: Bearing is a mechanical element that permits relative motion between two parts, such as the shaft and the housing, with minimum friction. The functions of the bearing are as follows:

- The bearing ensures free rotation of the shaft or the axle with minimum friction.
- The bearing supports the shaft or the axle and holds it in the correct position.
- The bearing takes up the forces that act on the shaft or the axle and transmits them to the frame or the foundation.

e) Pillow block bearing: A pillow block bearing is used in low-torque, light-load applications. With this configuration, the pillow block is bolted to a foundation, securing it, while the shaft and the inner ring of the bearing are free to rotate. With split pillow blocks, the housing element or cap can be separated from the base.

We will be using the type of block bearings so we can reduce friction created when our bench is in rotation. Also, it reduces the vibrations created by the rotating parts.

While the ball and cage itself can minimize friction, pillow block bearings also support the use of oil rings. An oil ring protects shafts from premature wear and tear by ensuring they are properly lubricating.
Pillow block bearings are easy to install. They can be mounted to the most stable surfaces using a supported type of fastener. Fasteners are driven through the pillow block bearing's sides and into the surface. This type of block helps us with the alignment of our system.

f) Inverter: An inverter controls the frequency of power supplied to an AC motor to control the rotation speed of the motor. Without an inverter, the AC motor would operate at full speed as soon as the power supply was turned ON. The use of an inverter to adjust the speed and acceleration of an AC motor increases the range of applications of the motor compared with a motor that operates at a constant speed. The acceleration rate is given as the change in speed over a specific period of time. We choose the Inverter Chnt NVF5-1.5.

5. Conclusion

It is the most important phase in any realization project, every element in the project is not randomly picked, and there are some processes, we went through to obtain a better solution and to act sustainably.

We started by setting the kinematic chain, and after that, we started the designing and sizing of our elements. For the shaft, we used the torsion, and flexion verifications. The results are satisfying. The motor and the reducer were chosen based on our needs and also based on the power and the general torque.

For the V-belt, we went through the whole process of sizing to obtain its length, number, and type of it. For the coupling, we mention the diameter of the pulley so we can set the wanted speed in the receiver (input speed of the reducer).

Finally, we had given an introduction about bearings and the type that we will be using also a small part about the inverter.
The logical continuation of this work is in [8].

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