

Bearing Extraction Press

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Abstract

This paper presents the design calculations and considerations for a bearing extractor tailored to remove deep groove ball bearings 6009, 6010, and 6011. The design addresses critical parameters such as loading, effective interference, interfacial pressure, and torque requirements. Key analyses are provided to ensure that the bearing extractor can safely and efficiently remove the specified bearings under typical operating conditions. The design also considers potential failure modes and includes appropriate adjustments to the crossbar and other components to prevent structural failure during operation.

Nomenclature

ϵ	Strain
μ	Friction Coefficient
σ	Stress (MPa)
τ	Torsion
B	Bore (mm)
C_{or}	Static Load Rating(N)
C_r	Dynamic Load Rating(N)
D	Outer Diameter (mm)
d	Inner Diameter (mm)
d_n	Nominal Diameter(mm)
E	Young's Modulus (GPa)
F_r	Radial Force
P	Pressure
p	Pitch
T	Torque (Nm)
W	Force (N)

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1. Declaration

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2. Introduction

Bearings play a crucial role in mechanical systems, and their removal often requires specialized equipment. This paper focuses on the design of a bearing extractor specifically for bearings 6009, 6010, and 6011. The objective is to develop an extractor capable of handling these bearings while addressing interference fit, loading conditions, and material properties. The design incorporates detailed calculations to ensure the extractor can operate under the required forces and torque, providing a reliable and safe solution for removing bearings without damaging surrounding components.

2.1. Objectives

- (i) To Design a bearing extractor for bearings 6009, 6010, and 6011, as described by the project brief.
- (ii) To provide calculations for the forces involved in the extraction of these bearings.
- (iii) To ensure that the bearing extractor will not fail.

2.2. Project Description

2.2.1. Parameters

- (i) Bearings 6009 - 6011
- (ii) Surface finish is ground.
- (iii) Load is 1940N.
- (iv) Temperature is 40 deg C.

2.2.2. Dimensions and Dynamic State Loading

Table 1 is an excerpt for the bearings requiring the extraction press.

3. Schematics

The schematic is in figure 2. Only examples of bearing 6009 and 6011 are shown because they are the extreme cases.

Bearing number	d	D	B	C_r	C_{or}
-	mm	mm	mm	N	N
6009	45	75	16	22000	15200
6010	50	80	16	22900	16600
6011	55	90	18	29700	21200

Table 1: Dimensions and Dynamic Loading of Given Bearings(1). See fig. 1

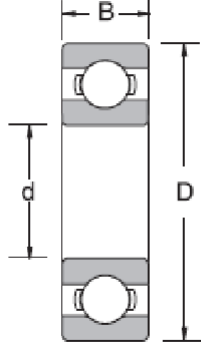


Fig. 1: Bearing Dimensions(1)

4. Calculations

4.1. Loading

When

$$F_r \leq 0.3 * C_{or} \quad (1)$$

then equation 2a is true. Otherwise 2b is true.

$$\Delta_{dF} = 0.08 * \sqrt{\frac{d * F_r}{B}}, \quad (2a)$$

$$\Delta_{dF} = 0.02 * \frac{x_r}{B}, \quad (2b)$$

4.1.1. Bearing 6009

$$0.3 * 15200 = 4560N, \quad (3a)$$

$$R_f = 1940 \leq 4560N, \quad (3b)$$

$$\Delta_{dF} = 0.08 * \sqrt{\frac{0.045 * 1940}{0.016}} = 5.909\mu m \quad (3c)$$

4.1.2. Bearing 6010

$$0.3 * 16600 = 4980N, \quad (4a)$$

$$R_f = 1940 \leq 4980N, \quad (4b)$$

$$\Delta_{dF} = 0.08 * \sqrt{\frac{0.05 * 1940}{0.016}} = 6.229\mu m \quad (4c)$$

4.1.3. Bearing 6011

$$0.3 * 21200 = 6360N, \quad (5a)$$

$$R_f = 1940 \leq 6360N, \quad (5b)$$

$$\Delta_{dF} = 0.08 * \sqrt{\frac{0.055 * 1940}{0.018}} = 6.159\mu m \quad (5c)$$

In table 2, there is a summary of the results of these calculations.

Table 2: Apparent Interference

Bearing Number	δ_{dF} (μm)
Bearing 6009	5.909
Bearing 6010	6.229
Bearing 6011	6.159

4.2. Effective Interference

Parameter g is between 1 and 2.5 micrometers on a ground shaft. Choose 2.5 micrometers as the worst case for interference.

Refer back to table 2 for the apparent interference. Effective interference is denoted by equation 6.

$$\Delta_{eff} = \Delta_{dF} + G \quad (6)$$

Therefore, using table 2 and this new found knowledge, effective interference can be assumed

4.2.1. Bearing 6009

$$5.909 + 2.5 = 8.409 \quad (7)$$

4.2.2. Bearing 6010

$$6.229 + 2.5 = 8.729 \quad (8)$$

4.2.3. Bearing 6011

$$6.159 + 2.5 = 8.659 \quad (9)$$

Table 3 has a summary of all these answers

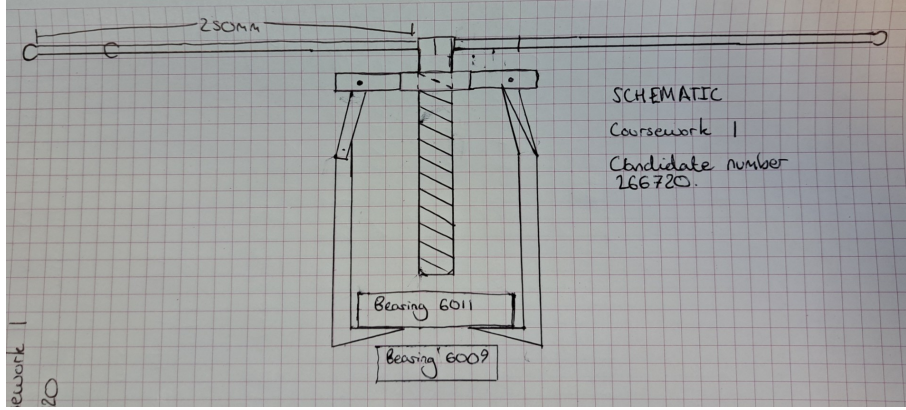


Fig. 2: Schematic

Table 3: Effective Interference

Bearing Number	Δ_{dF} (μm)
Bearing 6009	8.409
Bearing 6010	8.729
Bearing 6011	8.659

4.3. Effect of Temperature

Interference between inner rings and steel shafts decreases due to temperature rise (ΔT) from bearing rotation. The minimum required interference is calculated considering this effect:

$$\Delta_{eff} = 0.0015 * d_n * \Delta T \quad (10)$$

4.3.1. Bearing 6009

Minimum required effective interference:

$$\Delta_{eff} = 0.0015 * 45 * 40 = 2.7 \quad (11)$$

4.3.2. Bearing 6010

Minimum required effective interference:

$$\Delta_{eff} = 0.0015 * 50 * 40 = 3 \quad (12)$$

4.3.3. Bearing 6011

Minimum required effective interference:

$$\Delta_{eff} = 0.0015 * 55 * 40 = 3.3 \quad (13)$$

Taking the minimum required effective interference of all three the minimum required in total will be the greatest number, to ensure safety. Therefore minimum possible interference is 3.3 micrometers

4.4. Interfacial Pressure

Radial interference is half Δ_{eff} In Figure 3: $R_1 = 0$ because the shaft is solid R_2 is the nominal inner radius of the bearing R_3 is the outer radius of the inner ring of the bearing; obviously this is not constant through the thickness of the ring and bearing manufacturers do not give this information.

Steel has the greatest elastic modulus and for the worst case scenario an Elastic Modulus of 210GPa

$$p = \frac{\Delta_{eff} * E * (R_2^2 - R_1^2) * (R_3^2 - R_2^2)}{2 * R_3^3 * (R_3^2 - R_1^2)}, \quad (14a)$$

$$R_3 = \frac{1}{2} * (d + (\frac{D-d}{2}) * 0.33) \quad (14b)$$

4.4.1. Bearing 6009

$$R_3 = \frac{1}{2} * (0.45 + (\frac{0.3}{2}) * 0.33) = 0.24975, \quad (15a)$$

$$p = \frac{8.409 * E * (0.0506) * (0.06238 - 0.0506)}{2 * 0.01139 * 0.06238} \quad (15b)$$

$$p = 741 \text{ GPa} \quad (15c)$$

4.4.2. Bearing 6010

$$R_3 = \frac{1}{2} * (0.50 + (\frac{0.3}{2}) * 0.33) = 0.27475, \quad (16a)$$

$$p = \frac{8.729 * E * (0.0625) * (0.07549 - 0.0625)}{2 * 0.015625 * 0.07549} \quad (16b)$$

$$p = 630 \text{ GPa} \quad (16c)$$

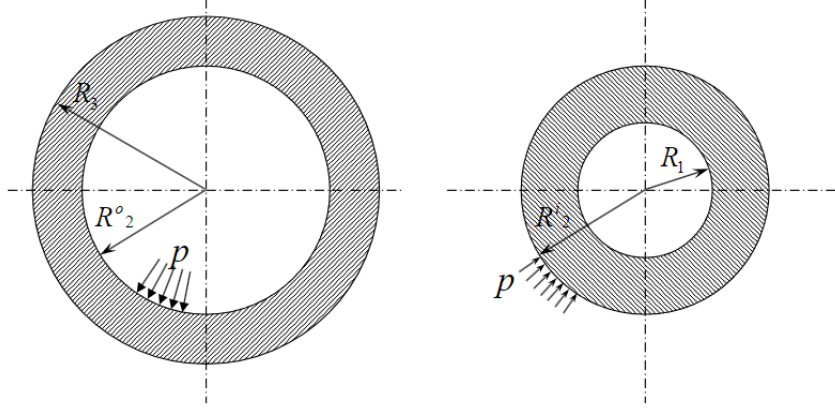


Fig. 3: Interfacial Pressure

4.4.3. Bearing 6011

$$R_3 = \frac{1}{2} * (0.55 + (\frac{0.35}{2}) * 0.33) = 0.303875, \quad (17a)$$

$$p = \frac{8.659 * E * (0.07425) * (0.09234 - 0.07425)}{2 * 0.02080 * 0.07549} \quad (17b)$$

$$p = 777 \text{ GPa} \quad (17c)$$

4.5. Force Required

See Table 4, for the static friction coefficients. From this table the static coefficient was assumed to be the worst case of all the materials; 1.35 for aluminium on aluminium.

$$W = PA\mu \quad (18a)$$

$$A = \pi dB \quad (18b)$$

Where $\mu = 1.35$

4.5.1. Bearing 6009

$$2.62 * 10^{-3} = \pi 0.045 * 0.016 \quad (19a)$$

$$W = 7.41 * 10^6 * 2.62 * 10^{-3} * 1.35 = 26,209 \text{ N} \quad (19b)$$

This clearly gives a number that is too large. Looking at the material table 4, and considering cheapest materials the choice is now cast iron and steel. This makes $\mu = 0.4$

$$2.62 * 10^{-3} = \pi 0.045 * 0.016 \quad (20a)$$

$$W = 7.41 * 10^6 * 2.62 * 10^{-3} * 0.4 = 7765.68 \text{ N} \quad (20b)$$

4.5.2. Bearing 6010

$$2.513 * 10^{-3} = \pi 0.05 * 0.016 \quad (21a)$$

$$W = 6.3 * 10^6 * 2.513 * 10^{-3} * 0.4 = 6332.76 \text{ N} \quad (21b)$$

4.5.3. Bearing 6011

$$3.11 * 10^{-3} = \pi 0.055 * 0.018 \quad (22a)$$

$$W = 7.77 * 10^6 * 3.11 * 10^{-3} * 0.4 = 9665.88 \text{ N} \quad (22b)$$

4.6. Torque Required

In Figure 4, there is all of the nomenclature visualised on a thread. The torque required to lift a load with a screw is in Equation 23.

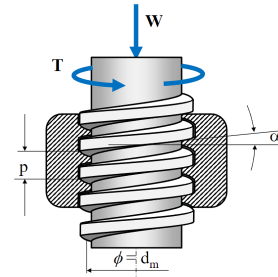


Fig. 4: Torque Required(1)

$$T = W * \frac{d_n}{2} * \frac{\sin \alpha + \mu \cos \alpha}{\cos \alpha - \mu \sin \alpha} \quad (23)$$

To determine the angle pitch must be chosen. This can be done using equation 24.

$$p = \pi * d_m * \tan a \quad (24)$$

Then rearrange to equation 28:

$$a = \arctan p / \pi * d_m \quad (25)$$

Now calculate for each bearing.

Material 1	Material 2	Dry surface	Greased Surface
Aluminium	Aluminium	1.05-1.35	0.3
Aluminium Bronze	Steel	0.45	-
Aluminium	Mild Steel	0.61	-
Brass	Steel	0.35	0.19
Brass	Cast Iron	0.3	-
Bronze	Steel	0.08-0.1	0.0004 - 0.06
Bronze	Cast Iron	0.13-0.22	0.05-0.08
Bronze-sintered	Steel	-	0.13
Cast Iron	Cast Iron	1.1	0.07
Cast Iron	Steel	0.4	0.21
Copper-lead alloy	Steel	0.22	-
Graphite	Steel	0.1	0.1
Phosphor-bronze	Steel	0.34	0.17
Steel	Steel	0.5-0.8	0.16
PTFE	Steel	0.05-0.2	-

Table 4: Static Friction coefficients of different material combinations(1)

4.6.1. Bearing 6009

$$0.036 - 0.006/2 = 0.033, \quad (26a)$$

$$a = \arctan 0.006/\pi * 0.033 = 3.3123, \quad (26b)$$

$$7765.68 * \frac{0.033}{2} * x = 167.3Nm, \quad (26c)$$

$$x = \frac{\sin 3.3123 + 1.35 \cos 3.3123}{\cos 3.3123 - 1.35 \sin 3.3123} \quad (26d)$$

4.6.2. Bearing 6010

$$6332.76 * \frac{0.033}{2} * \frac{\sin 3.3123 + 1.35 \cos 3.3123}{\cos 3.3123 - 1.35 \sin 3.3123} = 136.45Nm \quad (27)$$

4.6.3. Bearing 6011

$$9665.88 * \frac{0.033}{2} * \frac{\sin 3.3123 + 1.35 \cos 3.3123}{\cos 3.3123 - 1.35 \sin 3.3123} = 205.9Nm \quad (28)$$

The greatest is bearing 6011 which has a torque of 205.9Nm. The average human can put between 400-450N of force on a handle with one hand. Take the lower end of that and multiply it by two for two hands and the output on the handle is 800N. This means to turn the 200N, the handle must be at least 0.25m long.

4.7. Potential Failure Modes

The potential failure modes are written out in table 5.

5. Internal Force Diagrams

Internal Force Diagrams are in Figure 5 and 6 respectively.

5.1. Failure Calculations

5.1.1. Crossbar Bending

Maximum Bending Stress is denoted by equation 29a.

$$\sigma = \frac{M * \frac{h}{2}}{I} \quad (29a)$$

$$I = \frac{bh^3}{12} \quad (29b)$$

Calculations for bearing 6011 due to it requiring the most force:

$$8.33 * 10^{-10} = \frac{0.01 * 0.01^3}{12} \quad (30a)$$

$$2.9GPa = \frac{0.05 * 9665.88 * \frac{0.01}{2}}{8.33 * 10^{-10}} \quad (30b)$$

As the yield of steel is 600-800 MPa(2). This means that the crossbar will yield. Therefore the bar should be thickened. At 20mm instead the equations look like this

$$1.33 * 10^{-8} = \frac{0.02 * 0.02^3}{12} \quad (31a)$$

$$363MPa = \frac{0.05 * 9665.88 * \frac{0.01}{2}}{1.33 * 10^{-8}} \quad (31b)$$

This makes the stress lower than the yield stress. The crossbar will now have a cross section of 20mm*20mm.

5.1.2. Lever Bending

Maximum Bending Stress is denoted by equation 29a, which has already been quoted by this report.

Part	Loading Type	Failure Modes
Jaw	Tension, Bending	Fracture, Plastic Deformation
Lever	Bending	Fracture, Plastic Deformation
Crossbar	Bending	Plastic Deformation
Screw	Torsion	Fracture

Table 5: Failure Modes

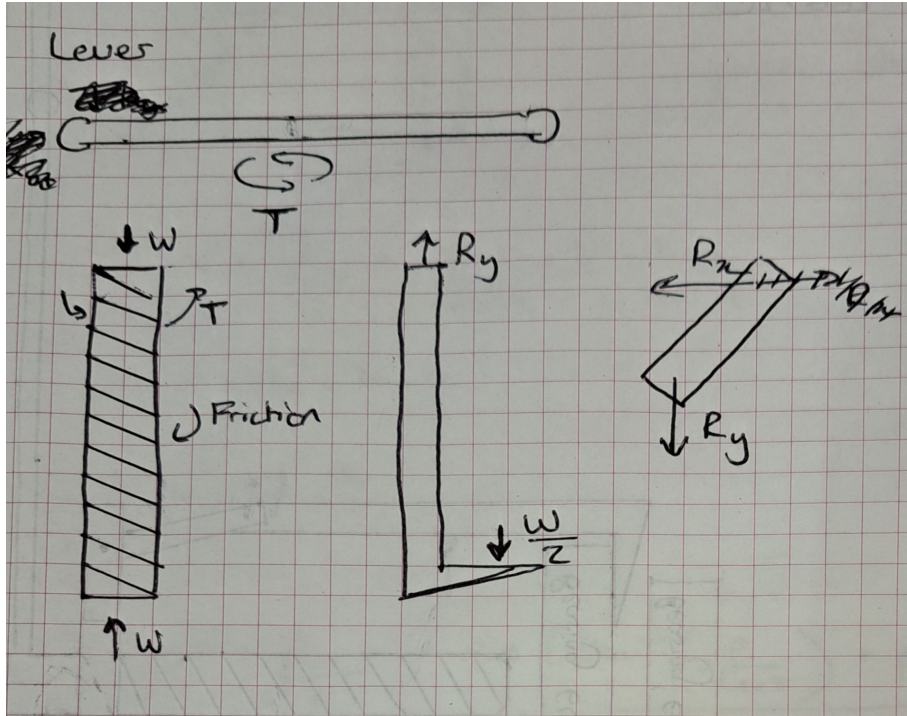


Fig. 5: Freebody Diagrams

However the inertia for cylinder cross sections is denoted by equation.

$$I = \frac{\pi * R^4}{2} \quad (32)$$

The starting diameter for the lever is 20mm, which means the radius is 10mm

$$1.57 * 10^{-8} = \frac{\pi * 0.01^4}{2} \quad (33a)$$

$$16.39MPa = \frac{0.25 * 205.9 * \frac{0.01}{2}}{1.57 * 10^{-8}} \quad (33b)$$

16.39MPa is under the yield stress so the lever is now the correct size.

5.1.3. Arm Tension

Due to forces involved, it may be prudent to remove the second arm section and make the idea simpler. The start point of the design should be 20mm*20mm.

$$\sigma = F/A \quad (34a)$$

$$9665.88/0.02^2 = 24.16MPa \quad (34b)$$

24.16 MPa is well under the yeild stress for steel. This makes it good to use. However, in order to cut costs and use less material, a smaller cross section may be permitted. Using 10mm*10mm instead, as shown in equation 35

$$9665.88/0.01^2 = 96.66MPa \quad (35)$$

Using 10mm*10mm gives a result of 96.66 MPa, which is still well under the yield stress, but uses only a quarter of the material, which makes it the superior choice.

6. Final Design

With all these calculations in mind and all the final dimensions put together, the final design is shown in fig. 7. Not visible in the figure is the lever dimensions,

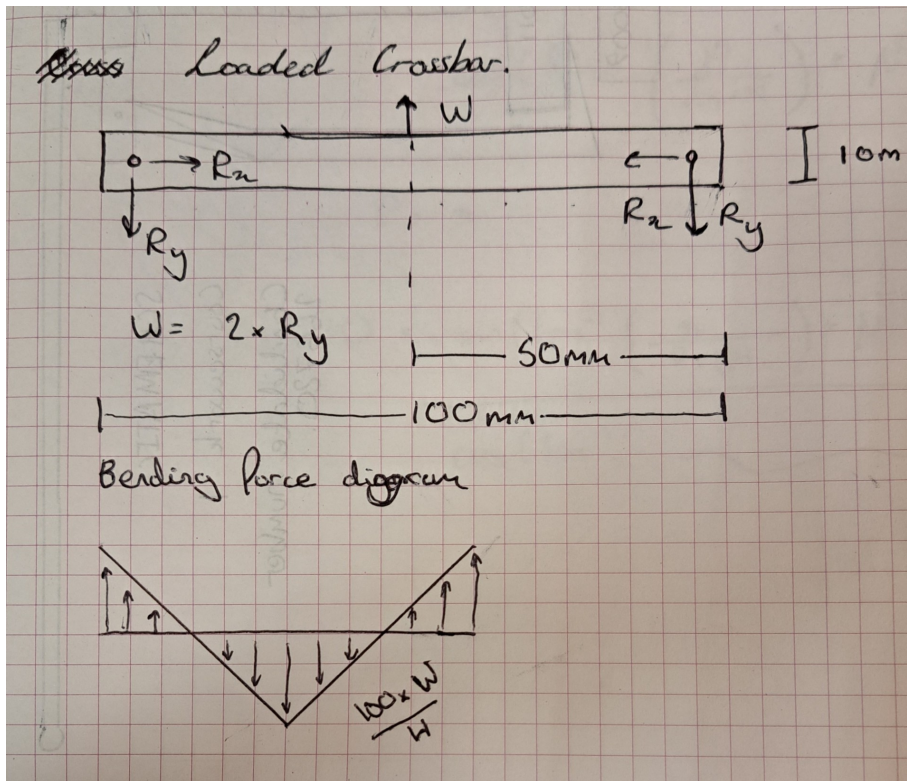


Fig. 6: Crossbar Bending

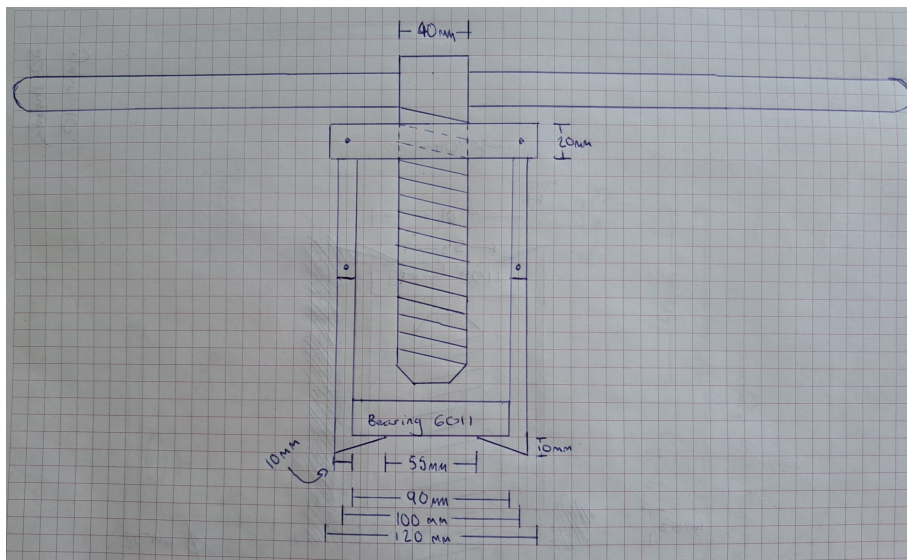


Fig. 7: Final Design

which are 250mm long and a 20mm diameter. The drawing is to scale but not lifesize.

It will be made of cast iron and steel

7. Conclusion

The bearing extractor designed for bearings 6009, 6010, and 6011 has been shown to meet the necessary operational requirements. Through a detailed examination of the loading, interference, interfacial pressure, and torque, the final design ensures sufficient strength and functionality. Key components, such as the crossbar, lever, and screw, have been reinforced to prevent failure. The final extractor design is robust, safe, and efficient for the specified bearings, ensuring ease of use in real-world

applications while minimizing the risk of material failure.

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