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, safe life methods, and stresses in critical areas. Key analyses are provided to ensure that the bearing extractor can safely and efficiently remove the specified bearings under typical operating conditions..

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)

Contents

1. Declaration

This report is submitted as part requirement for the degree of Mechanical Engineering at the University of Sussex. It is the product of my own labour except where indicated in the text. The report may be freely copied and distributed provided the source is acknowledged. I hereby give permission for a copy of this report to be loaned out to students in future years.

2. Introduction

The objective for this report is to create an intermediary design of the bearing extractor, expanding on the work done in week 4.

2.1. Objectives

(i) To choose a suitable thread and calculate stresses in the body of the screw and in the thread at three critical points.
(ii) To choose suitable materials
(iii) To calculate dimensions using a safe life design concept
(iv) Identify Stress Concentrators and calculate stresses at critical sections of two elements
(v) To create technical drawings of two elements of the device specifying tolerances and surface conditions
(vi) Draw the assembly drawing of the device specifying fits where needed.

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 5 is an excerpt for the bearings requiring the extraction press.

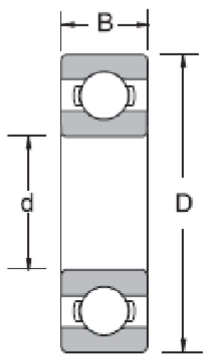

Fig. 1: Bearing Dimensions(1)

3. Thread Choice

The thread choice is a quite important decision to make. In this paper, the choice between square and trapezoidal will be assessed on 4 factors. These are load capacity, efficiency, manufacturability, and durability.

In terms of load capacity, square threads are ideal for high-load applications because they have a very high efficiency and minimal friction. They can handle higher axial loads compared to trapezoidal threads for the same size. However, trapezoidal threads are

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

also suitable for carrying loads, but they may not be as efficient as square threads due to increased friction. However, they are still good for moderate to high-load applications.(2)

Square threads are very efficient due to their low friction, especially useful in applications where you need to transfer torque to linear motion with minimal losses. Trapezoidal threads are slightly less efficient because of the thread profile, which increases friction.

Considering the manufacturing element, square threads are harder to machine because of the right-angle profile. They may require specialized tools or processes, increasing manufacturing costs. Trapezoidal Threads are easier to manufacture and more common, as they have a wider thread angle (usually 30°). They are often preferred for ease of production and availability of tools.

Square threads show less wear because of the direct axial force transmission, but they may not distribute wear as evenly as trapezoidal threads. Trapezoidal is more wear-resistant due to a larger contact area. This makes it suitable for applications where durability and long service life are critical.

So due to the needs of this paper, the thread choice is trapezoidal. Now the size must be chosen. This report has already decided on a nominal diameter in the section part dimensions, based on calculations within the last report and also mentioned later in this report, which is 40mm. Therefore, there are only 3 options for thread size that can be chosen: 40x3mm, 40x7mm, and 40x10mm.

In equation 1, the compressive force can be extracted.

$$P_{cr} = \frac{\pi^2 * E * I}{l^2} \tag{1}$$

The maximum compaction force seen in the previous report was 9665.88N and a safety factor of 1.5 is chosen. Therefore, the compressive force will be $P_{cr} = 9665.88 * 1.5 = 14498.82$N.

$$14498.82 = \frac{\pi^2 * E * I}{l^2} \tag{2a}$$

$$I = \frac{\pi * d_r^4}{4} = \frac{14498.82 * l^2}{\pi^2 * E} = 2.2667 * 10^{-8} m^4 \tag{2b}$$

$$d_r = 26.067mm \tag{2c}$$

In table 2, you can see the thread dimensions required. From this dimension table, this report can choose with confidence and a factor of safety of 1.5 the TR40*10mm.

For the external loading on the screw (i.e. load that needs to be moved), calculate the required torque applied.

$$T = W * \frac{d_m}{2} * \frac{l + \pi * \mu * d_m * sec\theta}{\pi * d_m - \mu * l * sec\theta} + \mu_c * W * R_c \tag{3}$$

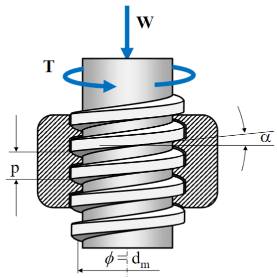

Fig. 2: Torque Required(1)

3.1. Stress in thread

The stress in the thread must be calculated at three critical points seen in Figure 3. In equation 24, τ is equal to $1.15 * 10^6$ and in equation 23, σ is equal to $11.53779 * 10^6$.

3.1.1. Stress at critical point A:

The stress diagram for critical point A is shown in figure 4. σ_b and is represented by the equation 4:

$$\sigma_b = \frac{6 * W}{\pi * d_p * n_t * p} = \frac{6 * 14498.82}{\pi * 0.029 * 3} = 31.8MPa \tag{4}$$

$$I_1 = \sigma_x + \sigma_y + \sigma_z, \tag{5a}$$

$$I_1 = -11.53779 * 10^6 = -11.53779MPa, \tag{5b}$$

$$I_2 = 0, \tag{5c}$$

$$I_3 = -\sigma_x * \tau_{yz}^2 = 0 \tag{5d}$$

$$\sigma_p^3 + 11.53779\sigma_p^2 + 0 = 0, \tag{6a}$$

$$\sigma_p = -11537790, \tag{6b}$$

Size mm	Designation -	Pitch mm	Major max mm	Major Min mm	Pitch Max mm	Pitch Min mm	Minor max mm	Minor min mm
40	TR40*3	3	40	39.764	38.415	38.165	36.500	36.103
40	TR40*7	7	40	39.575	36.375	36.020	32.000	31.431
40	TR40*10	10	40	39.470	34.850	34.450	29.000	28.350

Table 2: Dimensions and Dynamic Loading of Given Bearings(1).

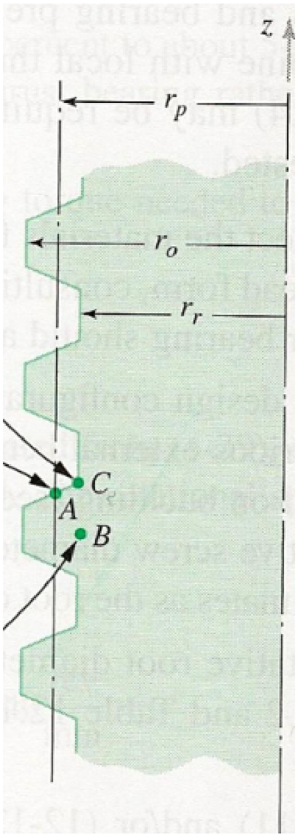

Fig. 3: Critical Points(1)

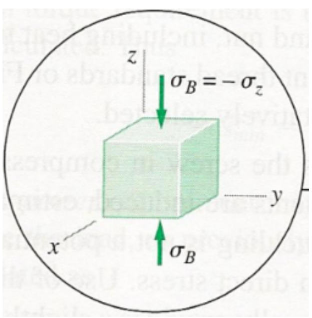

Fig. 4: Stress at critical point A(1)

There is only one real answer. This is under the value for steel and therefore no change to the design is required. This makes sense because there is only really one stress acting on A, the rest are more complicated.

3.1.2. Stress at critical point B:

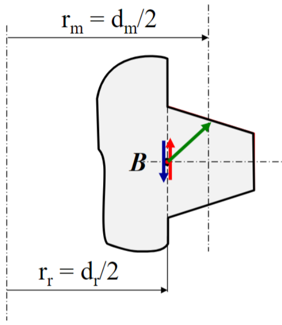

Fig. 5: Stress at critical point B(1)

The stress diagram for critical point C is shown in figure 6. The red arrow is τ which is equal to $1.15 * 10^6$, the blue arrow is σ which is equal to $11.53779 * 10^6$ and the pink arrow is σ_b and is represented by the equation 10:

$$\tau_x z = \frac{3 * W}{\pi * d_r * n_t * p} = \frac{6 * 14498.82}{\pi * 0.029 * 3} = 15.9MPa \tag{7}$$

$$\begin{aligned}
I_1 &= \sigma_x + \sigma_y + \sigma_z, && \text{(8a)}\\
I_1 &= 0 - 11.53779 * 10^6 = -11.539MPa, && \text{(8b)}\\
I_2 &= \sigma_x\sigma_y + \sigma_y\sigma_z + \sigma_x\sigma_z - (\tau_{xy}^2 + \tau_{yz}^2 + \tau_{xz}^2), && \text{(8c)}\\
0 &- ((1.15 * 10^6)^2 + (15.9 * 10^6)^2) = I_2, && \text{(8d)}\\
I_2 &= -17.05MPa && \text{(8e)}\\
I_3 &= -\sigma_x * \tau_{yz}^2 = 0 * 1.15 = 0 && \text{(8f)}
\end{aligned}$$

$$\begin{aligned}
&\sigma_p^3 + 11.539MPa\sigma_p^2 - 17.05MPa\sigma_p + 0 = 0, && \text{(9a)}\\
&\sigma_p = -9.8MPa, && \text{(9b)}\\
&\sigma_p = -1.74MPa && \text{(9c)}
\end{aligned}$$

This is under the value for steel and therefore no change to the design is required.

3.1.3. Stress at critical point C:

The stress diagram for critical point C is shown in figure 6. The red arrow is τ which is equal to $1.15 * 10^6$, the blue arrow is σ which is equal to $11.53779 * 10^6$ and the pink arrow is σ_b and is represented by the equation 10:

$$\sigma_b = \frac{6 * W}{\pi * d_r * n_t * p} = \frac{6 * 14498.82}{\pi * 0.029 * 3} = 31.8MPa \tag{10}$$

$$I_1 = \sigma_x + \sigma_y + \sigma_z, \tag{11a}$$
$$I_1 = 31.8 * 10^6 - 11.53779 * 10^6 = 20.3MPa, \tag{11b}$$
$$I_2 = \sigma_x\sigma_y + \sigma_y\sigma_z + \sigma_x\sigma_z - (\tau_{xy}^2 + \tau_{yz}^2 + \tau_{xz}^2), \tag{11c}$$
$$31.8 * 10^6 * -11.539 * 10^6 - (1.15 * 10^6) = I_2, \tag{11d}$$
$$I_2 = -366.85MPa \tag{11e}$$
$$I_3 = -\sigma_x * \tau_{yz}^2 = 31.8 * 1.15 = 36.57 * 10^{18} \tag{11f}$$

$$\sigma_p^3 + 20.3MPa\sigma_p^2 + 366.85GPa\sigma_p + 36.57TPa = 0, \tag{12a}$$
$$\sigma_p = -100239.79149, \tag{12b}$$

There is only one real answer. This is under the value for steel and therefore no change to the design is required.

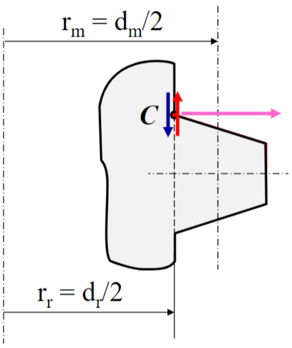

Fig. 6: Stress at critical point C(1)

4. Material Choice

Material choice was chosen to be cast iron and steel in the last report, due to the friction coefficient and manufacturing benefits. This report will check to ensure that these materials still make sense and which parts will be made of which material.

This report tentatively select 1020 Steel for screw, and A438-80 Cast Iron. These are selected initlially due to them being industry standard, cheap and having a relatively low friction coefficient.

5. Torque and Thread calculations

In Figure 2, there is all of the nomenclature visualised on a thread. The torque required to lift a load with a screw is in Equation 3.

To determine the angle pitch must be chosen. This can be done using equation 13.

$$p = \pi * d_m * \tan a \tag{13}$$

Then rearrange to equation 17:

$$a = \arctan p/\pi * d_n \tag{14}$$

Now calculate for each bearing. Pitch has already been chosen as 10mm.

5.0.1. Bearing 6009

$$0.036 - 0.006/2 = 0.033, \tag{15a}$$
$$a = \arctan 0.010/\pi * 0.033 = 3.3123, \tag{15b}$$
$$7765.68 * \frac{0.033}{2} * x = 167.3Nm, \tag{15c}$$
$$x = \frac{\sin 3.3123 + 1.35\cos 3.3123}{\cos 3.3123 - 1.35\sin 3.3123} \tag{15d}$$

5.0.2. Bearing 6010

$$6332.76 * \frac{0.033}{2} * \frac{\sin 3.3123 + 1.35\cos 3.3123}{\cos 3.3123 - 1.35\sin 3.3123} = 136.45Nm \tag{16}$$

5.0.3. Bearing 6011

$$9665.88 * \frac{0.033}{2} * \frac{\sin 3.3123 + 1.35\cos 3.3123}{\cos 3.3123 - 1.35\sin 3.3123} = 205.9Nm \tag{17}$$

The greatest is bearing 6011 which has a torque of 205.9Nm.

The efficiency of this screw is:

$$e = \frac{cos(3.3123) - 0.4 * tan(3.3123)}{cos(3.3123) + 0.4 * cot(3.3123)} = 0.123468 \tag{18}$$

The stresses in the body due to main loading can now be calculated. Maximum shear stress at the root of the screw:

$$\tau = \frac{T * d_r}{2 * I_p} = \frac{16 * 14498.82}{\pi * 40^3} = 1.153779 * 10^6 Pa \tag{19}$$

For axial stress in the body of the screw the force is needed. This comes from calculations for each bearing.

The μ for Steel and Cast Iron is $\mu = 0.4$ as seen in table 3

Therefore, for each bearing here are the equations:

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 3: Static Friction coefficients of different material combinations(1)

5.0.4. Bearing 6009

$$2.62 * 10^-3 = \pi 0.045 * 0.016 \tag{20a}$$
$$W = 7.41 * 10^6 * 2.62 * 10^-3 * 0.4 = 7765.68N \tag{20b}$$

5.0.5. Bearing 6010

$$2.513 * 10^-3 = \pi 0.05 * 0.016 \tag{21a}$$
$$W = 6.3 * 10^6 * 2.513 * 10^-3 * 0.4 = 6332.76N \tag{21b}$$

5.0.6. Bearing 6011

$$3.11 * 10^-3 = \pi 0.055 * 0.018 \tag{22a}$$
$$W = 7.77 * 10^6 * 3.11 * 10^-3 * 0.4 = 9665.88N \tag{22b}$$

The highest is 9665.88 which multiplied by 1.5 is 14498.82.

$$\sigma = \frac{W}{A} = \frac{4 * 14498.82}{\pi * 40^3} = 11.53779 * 10^6 Pa \tag{23}$$

The bearing stress on the thread is:

$$\sigma_B = \frac{2W}{\pi * d_m * p * n_t} = \frac{2 * 14498.82}{\pi * .040 * 3} = 7.69186 * 10^6 Pa \tag{24}$$

The n_t is usually 3 from practice, because usually only 3 threads hold the load. In table 4, it shows using the materials that have been chosen what safe pressure can be used.

These points are plotted in figure 7, with a power regression line derived from the experimental points provided, the purple representing the safe operating area and the green line representing the result we obtained. What the plot shows, assuming that the intrapolation is correct, is that the safe operating speed of the design is a maximum of 0.093317m/s.

6. Part Dimensions

The part dimensions are as follows:

This report will go through a quick summary of why these dimensions will with withstand the force, but for the full reasoning see the preliminary design report which is in the appendix.

The part dimensions are also put together in the technical drawings that are also in the appendix.

6.1. Screw

The screw is 40mm in diameter because of calculations earlier in this report.

6.2. Lever/Handle

The lever/handle was calculated with a 20mm diameter. Here are the calculations:

$$I = \frac{\pi * R^4}{2} \tag{25}$$

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

$$1.57 * 10^{-8} = \frac{\pi * 0.01^4}{2} \tag{26a}$$
$$16.39MPa = \frac{0.25 * 205.9 * \frac{0.01}{2}}{1.57 * 10^{-8}} \tag{26b}$$

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

Screw Material	Nut Material	Safe Pressure	Speed
-	-	MPa	m/s
Steel	Cast Iron	4.1	0.2032
Steel	Cast Iron	6.9	0.1016
Steel	Cast Iron	17.1	0.0406

Table 4: Safe operating pressure(2).

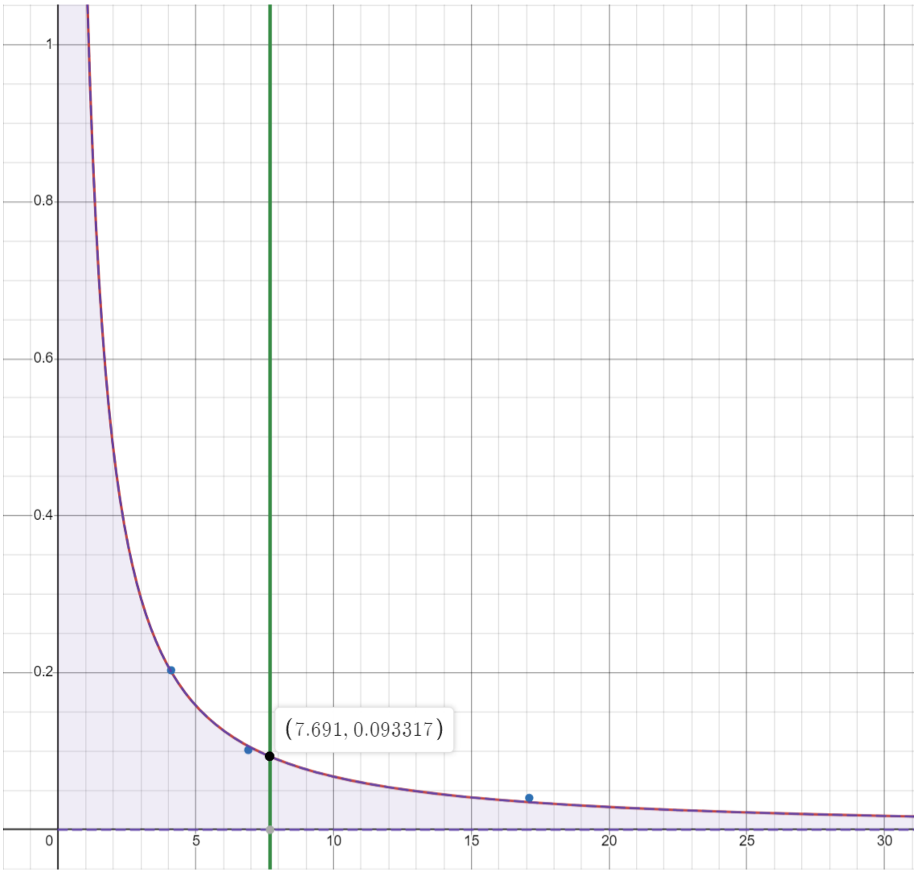

Fig. 7: Safety region plot

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

6.3. Arm

The arm was originally tested at 20mm*20mm cross sectional area but the calculations showed that a lot smaller arm was permitted. Therefore, calculations for a 10mm*10mm were performed an it passed, so, this means that less material has to be used. Here are the calculations:

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.

$$\sigma = F/A \tag{27a}$$

$$9665.88/0.02^2 = 24.16MPa \tag{27b}$$

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 28

$$9665.88/0.01^2 = 96.66MPa \tag{28}$$

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.

7. Stress Concentrators

7.1. Arm

The arm, made of steel, has had analysis done on it, and the maximum stress is $2.71 * 10^8$ which is under the yield stress of steel at $3.516 * 10^8$. Figure 8 shows the concentration of stress.

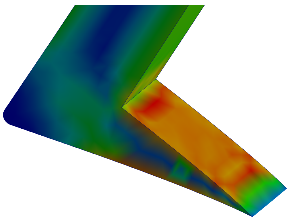

Fig. 8: Arm Stress Concetration

7.2. Crossbar

The crossbar, made of cast iron, has had analysis done on it and the maximum stress is $2.272 * 10^8$ and the yeild strength is $2.575 * 10^8$. Figure 9 shows the concentration of stress.

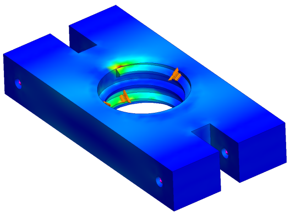

Fig. 9: Crossbar Stress Concetration

7.3. Screw

The screw, made of steel, has had analysis done on it, and the maximum stress is $2.71*10^8$ which is under the yield stress of steel at $3.516 * 10^8$. Figure 10 and 11 shows the concentration of stress. The top has some stress around where the handlebar/lever goes through it.

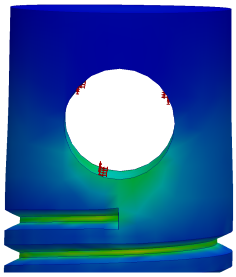

Fig. 10: Screw Stress Concetration

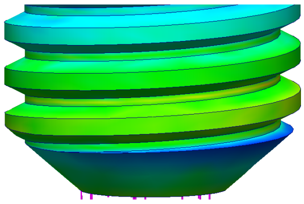

Fig. 11: Screw Stress Concetration

Every part has passed a safe life test with forces that are 1.5 times the size of the maximum force the extractor will experience even with the worst bearing.

8. Technical Drawings

Technical Drawings of the crossbar, arm and screw are attached to this report.

9. Assembly Drawing

The assembly drawing is attached to this report as well.

10. Conclusion

The intermediary design shows that the ideas in the first report make sense. The crossbar will be made of cast iron and the rest will be made of steel. The thread will be a TR40*10 which has been calculated to be the best for this project. The stress has been calculated at three critical points. The material has been chosen as 020 Steel for screw and A438-80 Cast Iron and has passed all the stress tests. These two materials consistently have a yield strength greater than the stress of the object. The part dimensions, technical drawings and assembly drawings have been included. More than two parts have been calculated using the safe liufe concept using a force that is 1.5 times the force this bearing extractor will encounter.

REFERENCES

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[2] Budynas, R.G., Nisbett, J.K. and Shigley, J.E. (2020). Shigley's mechanical engineering design. 11th ed. New York, Ny: Mcgraw-Hill Education.