Durham University

Design Report

Group 25
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1 Executive Summary

Microphonic effects exist in Hi-Fi systems wherein mechanical vibrations are transformed into unwanted electrical signals, termed noise, that consequently reduce sound quality. Isofonics have designed footers of aluminium and steel upon which such systems are to be supported that aim to isolate and attenuate the signals responsible for this aforementioned noise. Three footers are required per system, three being the minimum number of points of contact necessary to support a rigid mass, and are to be sold as a device comprising a contained magnetic damping mechanism with two optional additional pieces: a removable base to use the device as a stand alone mount as opposed to its mounting within existing equipment and a removable spike offering a minimal point of contact. The main device, base and spike are to cost £blah, £blah and £blah respectively, placing Isofonics at the higher end of the existing market due to its premium quality components embodying a design that is both innovative and effective. The device may be altered manually by the user to support a range of different masses, a feature of customisability unique to Isofonics. This process may also be informed by a user friendly phone application. Additionally, within this design lies potential for product families of differing size, catering to an even wider audience, and applications within the field of optics, more specifically in the optimisation of microscopes.

Contents

2 Introduction

Microphonics describes the phenomenon wherein internal components within an electronic device transform mechanical vibrations into undesired electrical signals[1]. In the context of hi-fi systems, and when these vibrations are within the frequency range audible to the human ear—20 Hz–20 kHz—they equate to noise that reduces audio quality and therefore threatens the user's listening experience. This report details the design of vibration isolation and attenuation mounts that minimise this interference as well as that originating from external sources.

The following function analysis tree defines the user requirement specification for the product:

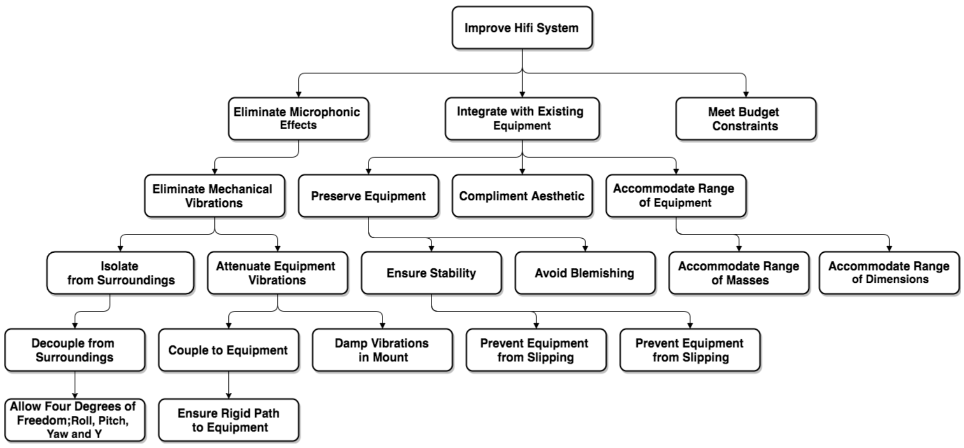


Figure 1: User Requirement Specification Diagram

With such a niche product consisting of high tolerance machined parts and premium materials comes a high cost, indicating a corresponding market; middle aged/mature clientele with strong technical understanding of the hobby and its principles or affluent younger hobbyists. The current market offers a range of mounts at a range of prices within which our product shall lie at the higher end, namely between £1500–£2000 per mount.

Market leaders such as Stillpoints®, Nordost® and Isoclear® offer sleek solutions yet lack the tailorability offered by our design, allowing customers to pre-load their mounts manually for individual amps, potentially with interactivity provided by an instructive technical app. After conducting market research via surveys, it was found that potential customers were very much interested in the ability to manually adjust their mount as appropriate and that this offered an invaluable, unique selling point.

This report covers the initial design concepts proposed by Group 25, the development of a chosen design, its design for manufacture and sustainability and its commercial considerations.

3 Design Concepts

During initial research it was noticed the most successful products on the market met three primary criteria:

- They allowed the mounted equipment to move with four degrees of freedom: three in translation and one in rotation.

- They used single degree of freedom systems for damping. Simpler damping mechanisms were more effective at consistently attenuating unwanted frequency components.
- Vibrations were efficiently transmitted to the damping system from the equipment. This was typically achieved using hard materials and limited contact between the footer and equipment.

In summary, any concepts generated had to first *isolate* the equipment then *attenuate* vibrations by allowing the equipment to move freely and then damp vibrations in controlled manner.

In order to simplify the process of concept generation, the formulation of isolation and attenuation mechanisms was separated. Isolation mechanisms were designed to translate the four degrees of freedom in the movement of the equipment into the single degree of freedom of the damping mechanism. Those single degree of freedom damping mechanisms were designed to be easily applied to multiple isolation concepts.

3.1 Damping Mechanisms

Three methods of damping were considered—not including the friction in bearings, which cannot be completely eliminated. These were viscoelastic, magnetic and fluid damping.

Viscoelastic materials, such as rubber or Sorbothane®, heat up when subjected to changes in mechanical stress [2]. Hence, the amplitude of vibrations are reduced when the driving oscillations are transmitted through viscoelastic materials. Typically when these materials are used in footers, the vibrations are not limited to a single degree of freedom, so the effectiveness of attenuation depends on the vibration direction. This makes it difficult to model and specify a single system natural frequency or damping factor.

Figure 2 details how viscoelastic materials can be used in a system where vibrations have been translated to a single direction. A linear bearing compresses a viscoelastic sleeve on the same shaft.

Magnets can also be used to damp vibrations when constrained to a single direction. As a permanent magnet travels through a coil, an EMF is induced due to Faraday's law, which drives a current when the circuit is complete. The current flowing generates a magnetic field opposing the permanent magnet, according to Lenz's law. A possible assembly for this system is detailed in Figure 3 on the following page. The strength of magnets were to be tuned to the desired dynamic properties of the system.

VISCOELASTIC DAMPING CONCEPT

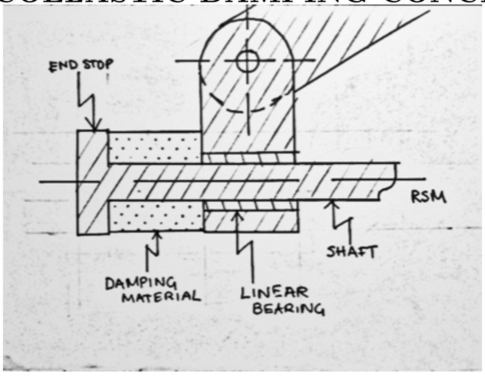


Figure 2: D1—Viscoelastic damping mechanism.

MAGNETIC DAMPING CONCEPT

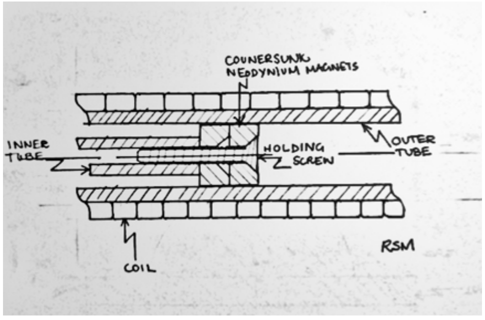


Figure 3: D2—Magnetic damping mechanism.

Finally, a simple fluid damping system was considered, whereupon a bearing travels through some viscous fluid. Skin friction provides the damping force in Figure 4 on the next page. The size of the bearing could be tuned to meet the required constraints for damping factor and natural frequency. However, this concept was difficult to realise in any manufacturable assembly whilst successfully containing the fluid.

3.2 Isolation Mechanisms

Early concepts made use of linkages to transform vertical and lateral oscillations into lateral oscillations along damped linear bearings, which would have been most easily realised using a viscoelastic damper. Figure 5 on the following page details one such embodiment.

Some components of vertical vibrations were not damped by the mechanism in Figure 5; these were transmitted to the base of the mount such that the mount was not completely isolated. Repeating the mechanism by using multiple layers of linkages was explored in the concept detailed in Figure 6 on page 7. Each layer damps a fraction of the vertical oscillations transmitted

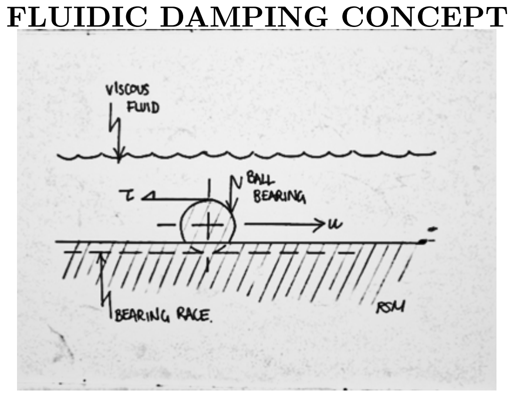


Figure 4: D3—Fluidic damping mechanism.

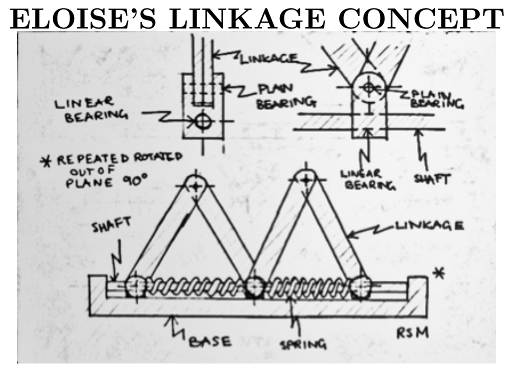


Figure 5: C1—Simple linkage mechanism.

downwards by the layer above resulting in more isolation.

However, the fundamental issue with linkage based isolaton mechanisms is the feature size of those linkages. Thin members are prone to resonance with large amplitudes at relatively lower frequencies **citation needed** possibly within the audible spectra (20 Hz–20 kHz).

Spherical bearings also featured in design concepts. Figure 7 on the following page describes one such embodiment where a layer of spherical bearings were sandwiched between two platforms. The upper platform was to be made of a hard material such as stainless steel or tungsten carbide, in order to efficiently transmit vibrations through to the bearings via point contacts. The bearings allowed the top platform to move with the equipment in three degrees of freedom: two lateral directions and lateral rotation. The bottom platform would have been made of some viscoelastic material with holes for the bearings to rest in. Both vertical and lateral oscillations would have been damped by this platform and dissipated as heat.

This concept had a major drawback: the damping mechanism was not confined to a single degree of freedom. It would have been challenging to

RUSSELL'S LINKAGE CONCEPT

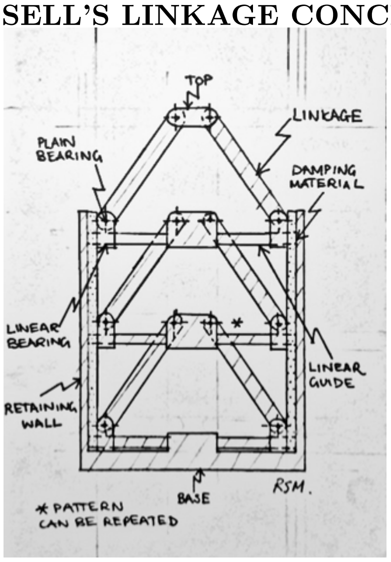
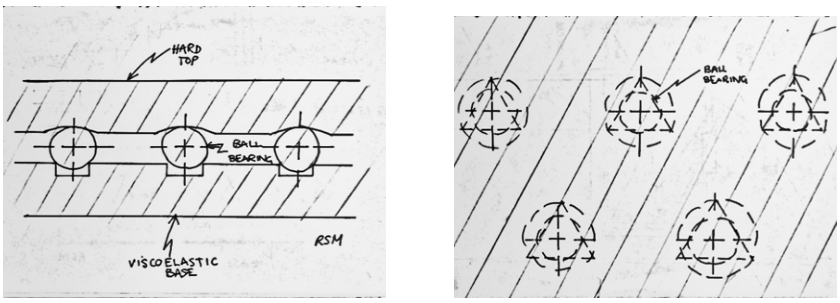


Figure 6: C2—Layered linkage mechanism.

determine a single damping factor and resonant frequency. The isolation mechanism was an anomaly in the concept generation phase, as the damping mechanism was not interchangable with those in Section 3.1 on page 4.

GEORGE'S PLATFORM CONCEPT



(a) Cross Section (b) Bird's Eye Cross Section

Figure 7: C3—Viscoelastic platform concept.

Isolation cones are a popular type of mount, an example is Nordost's Sort Kones®. The Sort Kones are made of three parts: a post, spherical bearing, and cone. The bearing rests on the top of the post and the cone sits on top of the bearing, such that the cone is free to swivel and rotate. However, cones seek only to isolate the mounted equipment and allow free movement—they do not attenuate vibrations.

Figure 8 on the following page demonstrates how an attenuating mechanism could have been incorporated into the post of an isolation cone. The spring in the isolation mechanism could have been replaced with one of the damping mechanisms in Figure 2 and Figure 3.

MAT'S CONE CONCEPT

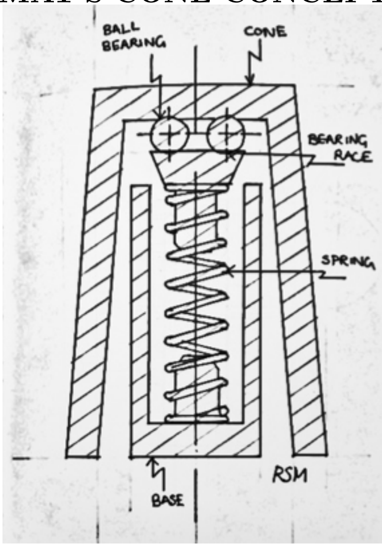


Figure 8: C4—Damped cone concept.

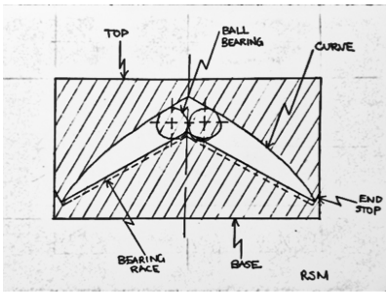
An alternative mechanism was devised to translate horizontal and lateral oscillations into oscillations of a spherical bearing along a race. Figure 9 on the next page details the geometry required to achieve this translation.

When no force is applied to the top piece, the bearing would travel to the bottom of its race. Increasing the load would cause the bearing to move up the race due to the gradient of the curve on the top piece, which was greater than the gradient of the race. There would be some component of the reaction force acting on the bearing perpendicular to the curve which points up the the slope of the race. However, the gradient of the curve decreases travelling up the slope, so for a given load the component of the reaction force would also decrease. Therefore, for each load, there is a different equilibrium position somewhere along that race where the components of the bearing's weight and the reaction force acting on the bearing in the direction of the race are equal and opposite. The mechanism acts as a geometric spring with some stiffness.

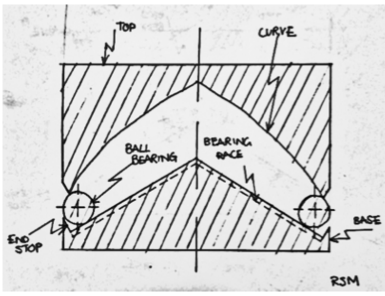
The frictional sheer forces in the mechanism would provide significant damping due to the magnitude of the reaction forces involved. Furthermore, one of the damping mechanisms in Figure 2 or Figure 3 could be coupled to the linear motion of the bearings to tune the system to a desired natural frequency and damping factor—this could result in quite a large footer.

A drawback of this concept is difficulty in containing the bearings, the user would be free to remove the top piece, and the bearings would all come out of their races. Competitors could easily see how the mechanism works

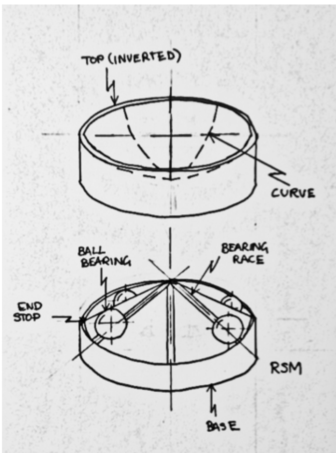
RUSSELL'S POINTY BOTTOM CONCEPT



(a) Cross Section- Under Load



(b) Cross Section- Under No Load



(c) 3-Dimensional

Figure 9: C5—Cone and bearing concept.

and replicate it for themselves.

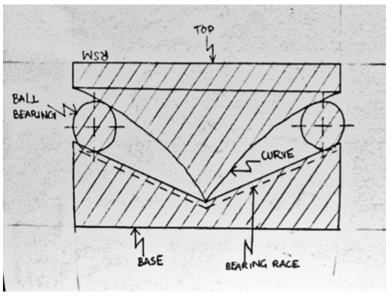
Figure 10 on the following page details a variation of C5, whereby the bearings travel outwards up the slope. This creates more space for a damping mechanism for each race around the outside of the footer. A fluid could be contained in the cavity in the bottom piece to damp vibrations using the mechanism described in Figure 4.

The flexibility of the alternative cone and bearing concept C6, coupled with its innovative design set the concept apart from its alternatives. In [3], it was shown how concept C6 with damping mechanism D2 best matched the customer's expectations and engineering requirements. This is the concept which informed the next stage of development and our final design.

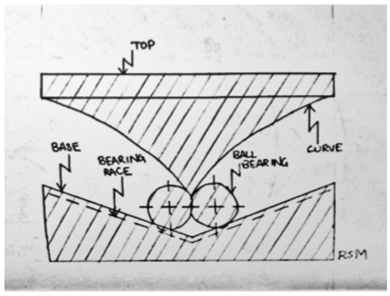
4 Design Development

Through an iterative process, the concept highlighted in [3] for its promise was increasingly refined to produce a final design. The key milestones in

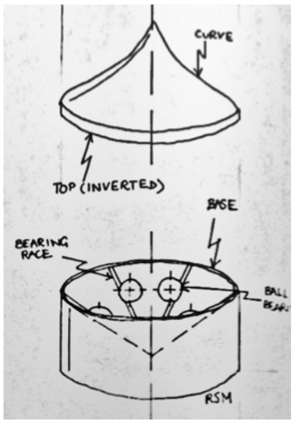
RUSSELL'S POINTY TOP CONCEPT



(a) Cross Section- Under Load



(b) Cross Section- Under No Load



(c) 3-Dimensional

Figure 10: C6—Alternative cone and bearing concept.

development are as follows:

- The transition from curved geometry to opposing pairs of rare earth magnets to support the load of equipment.
- The addition of a user-adjustable preloading mechanism to increase the range of masses that could be supported to full mass range of amplifiers: 5 kg–30 kg **citation needed**
- The casing required to contain the isolation mechanism were considered.

4.1 Opposing Magnets

4.1.1 Magnet Simulation

As the footer houses 12 rare earth magnets, considerations have to be made for the way in which these magnets interact with the surroundings of the footer as they could cause damage or interfere with mounted or surrounding equipment footer. Using a software package called MagNet® produced by infolytica®, it was possible to perform a 2D finite element analysis of the footers, allowing the effects of these magnets to be modelled. The analysis performed in this section is for the geometry of the final design, but the same procedure was performed following each significant revision during the design process. When modelling the effect of these magnets the two extreme cases were considered: at minimum and maximum separation as detailed in Figure 11 and Figure 12.

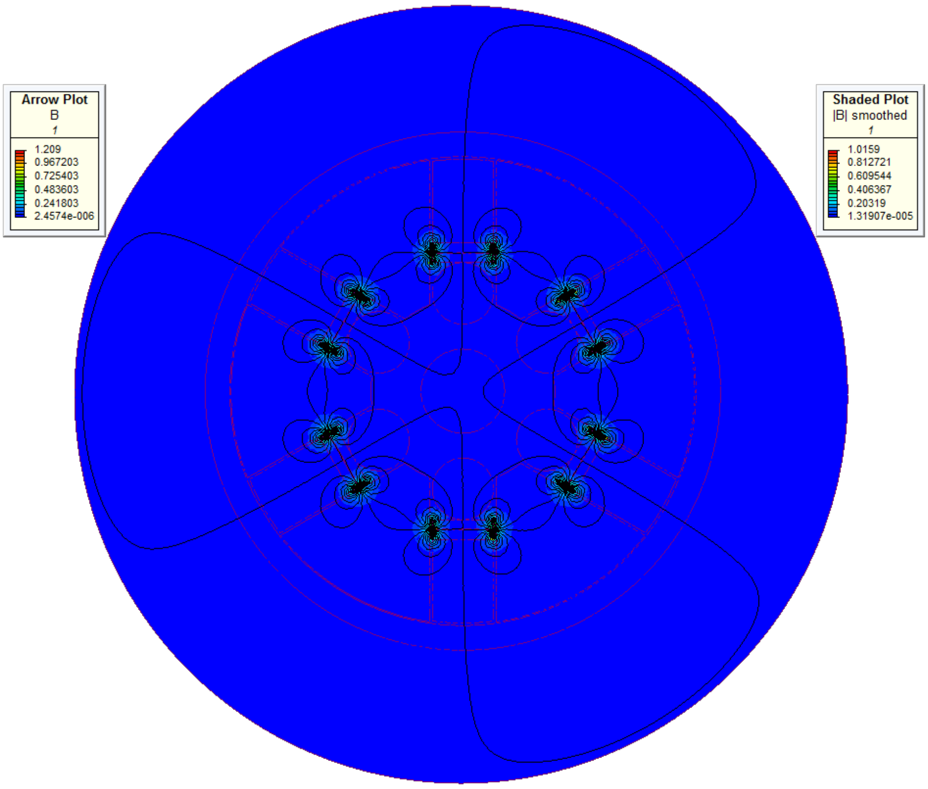


Figure 11: Flux state with minimum separation of magnets

It can be seen that the flux density outside of the footer is negligible as it has a value of the order 10^{-5} T meaning that there would be minimal to no interactions between the footer and its surroundings as nearly all of the magnetic flux is contained within the core of the device.

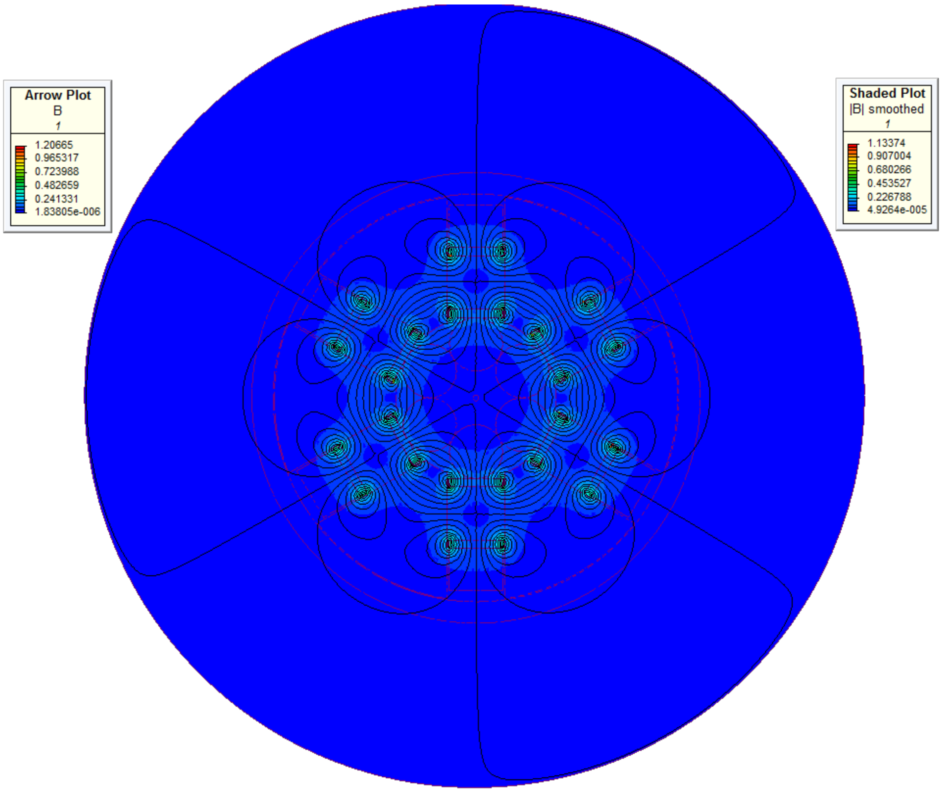


Figure 12: Flux state with maximum separation of magnets

4.2 Preloading Mechanisms

4.3 Casing

4.4 Materials

4.5 Materials

Neodymium magnets are the most widely used type of rare-earth magnet, are the strongest type of permanent magnet commercially available and are of greatest benefit in applications where there is limited space. The magnetic properties termed remanence, coercivity and energy product dictate the performance of neodymium magnets. They measure the strength of the magnetic field, the material?s resistance to becoming demagnetized and the density of magnetic energy respectively. Given that there are numerous grades available, an overview of the range of values are outlined in Table 1. [1] The grade chosen for the project was N45SH, coated with three layers (nickel, copper and nickel) to reduce corrosion and provide a smooth finish. The brittleness of the magnets was also compensated for such that it does not undergo shear force during the cyclic loading.

Tungsten carbide possesses Young?s modulus 530 GPa, more than two times of that for the grade of stainless steel chosen (190 ? 203 GPa) [4] [5]. Due to Hooke?s law, the material does not undergo large deflections which in turn ensured efficient energy transfer between the Hi-Fi components. The bearings do not require any lubrication as they grind down any particulates that may enter the system.

Steel is harder, and therefore more expensive, to machine than softer materials however, its properties make it invaluable to the quality of the product (see Section 5) and pose no problem for the advanced machining capabilities of today. The product's entire composition out of steel was initially proposed but after receiving a quote from 'Sylatech' (r/c), a machining company in the northeast of England for £—- per piece, it was evident some optimisation was required to quote a retail price for our product that allowed for a 35% [4] profit margin.For this reason, it was decided that all non-critical parts may be machined from aircraft grade aluminium (6061-T6), an alternative roughly 1.5 times cheaper[5] than steel. 'Non-critical' in this context refers to all parts excluding the complex central piece (COR080-0003), the top piece (TOP080-0004) and removable spike (USR080-0001). For these parts, steel is required to contain magnetic flux leakage, remain unaffected by frictional effects of the moving bearings and not yield under concentrated stress. Additionally, aluminium is roughy 5 times easier to machine than stainless steel[5] saving significantly on time, and therefore cost.

4.5.1 Static Analysis

4.6 Dynamic Analysis

4.6.1 Parametric Model

4.6.2 System Tuning

4.6.3 Experimentation

To assess the effectiveness of the product, acceleration data from a vibrating amplifier was recorded to determine the frequency content of the signal to be attenuated. The same setup was used to capture acceleration data from the amplifier when mounted on half squash balls, which was later compared to the output of a dynamic model representing the final mount design using the undamped acceleration data captured as the model input. Squash balls were used to isolate Hi-Fi systems before footers entered the market and are an entry level alternative to footers.

A USB PicoScope[1] was used to capture the output of an accelerometer[2] at a sample rate of 50 kHz for 10 s. Human hearing ranges from 20 Hz–20 kHz;

[1]Pico Technology PicoScope 2204A
[2]STMicroelectronics LIS344ALH

according to Nyquist, the sample rate had to be at least twice 20 kHz to determine the power of these frequency components.

The accelerometer was coupled to the top of an amplifier connected to a standard AC mains power supply—nominal voltage of 240 V at a frequency of 50 Hz. Data from the accelerometer was captured for three conditions: with the amplifier turned off, on, and the amplifier turned on whilst sitting on half squash balls.

The data captured came in the form of a voltage output. To determine the acceleration from this information it was necessary to form a transfer equation. To do so, the technical data sheet for the accelerometer [6] was found and in it there was clear conversion which lead to the following

$$\ddot{y} = 9.81 \cdot (0.66 \cdot V - 1.65) \tag{1}$$

where a is the acceleration in $\mathrm{m\,s^{-2}}$ and V is the voltage measured in V. Equation 1 is linear, so has no affect on the shape of the spectra. The spectra of the measured voltages were analysed instead.

Firstly, the data for the amplifier when off—the control—was compared to the amplifier when on; their spectra up to 1 kHz are shown in Figure 13. There were no observable components above the noise floor in the range 1 kHz–25 kHz.

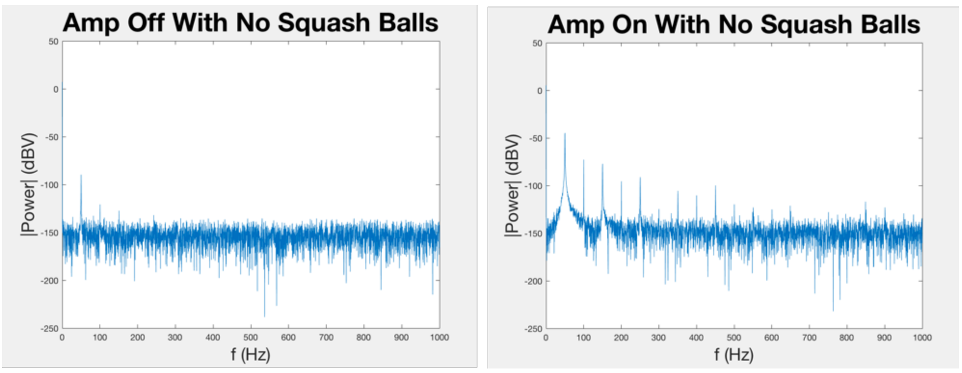


Figure 13: Compared Frequency Responses when Amplifier off and on

The spectra for amplifier when off had a fundamental at 50 Hz. This is likely due to interference from other mains powered devices in the room—air conditioning, PCs and desktop power supplies for example. Contrasting this control spectra to the spectra for the amplifier when turned on, the fundamental frequency increased in power significantly from −90 dBV to −40 dBV—a factor of 10^5. The harmonics can be seen at 50 Hz intervals although the noise floor due to the error of the accelerometer remained at −140 dBV.

Next, the spectra for the amplifier when on was compared to the spectra for the amplifier damped using squash balls. Figure 14 on the following page reveals the effect the squash balls had on the frequency content of observed vibrations.

The fundamental remains constant in frequency and power. However, from the simple damping the squash balls provide, the noise has been significantly cleared up, along with this success the harmonics have consistently lower peaks. This provides an example of successful damping which can be used as a comparison to the later mathematically damped data using dynamic model. The aim is to further clean up the noise and lower the power of the harmonics as much as possible, preferably below the noise floor which should lead to a reduction in microphonic effects and increase in sound quality.

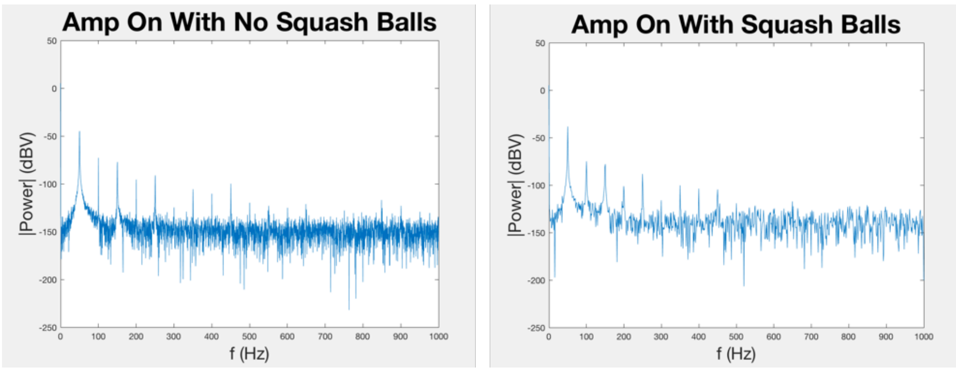


Figure 14: Compared Frequency Responses with and without Squash Balls

4.6.4 Dynamic Simulation

It was determined that a dynamic model of the system would be necessary to quantify the potential success of the product. This could visually display the effects of the footers on attenuation of a signal from a Hi-Fi system, and hence inform whether the sound quality would be improved.

The three footers and mounted equipment can be simplified to a spring-mass damper system with non-linear stiffness—a driving force F_d is supplied by the vibration of the Hi-Fi system, and the opposing forces from the magnetic repulsion force $F_m(y)$, damping force $c\dot{y}$ and system inertia $m\ddot{y}$. Considering these components, it was possible to develop a second order differential equation which when solved would characterise the effectiveness of the footers.

$$F_d = m\ddot{y} + c\dot{y} + F_m(y) \tag{2}$$

Simulink® was chosen to drive the dynamic model due to the straightforward construction of differential equations it provides; it offers a visual representation of the system and works hand-in-hand with MATLAB allowing for simple data processing. Simulink computes an iterative numerical solution and outputs a corresponding time series to the MATLAB workspace which can be transformed into the frequency domain for analysis.

Four components needed to be defined to complete the model:

- The mass of the supported equipment and top piece m. The simulation was run on the nominal supported mass of 20 kg.
- Constant c, which is dependent on the supported mass and the frictional coefficient μ between the tungsten carbide balls, and stainless steel cones and races for all three footers.
- A transfer function $F_m(y)$ to find the magnetic repulsion force given the vertical displacement of the equipment.
- F_d which varies with time and the input to the system, found by applying Newton's second law to the undamped accelerations measured in Section 4.6.3.

Section 4.6.1 details how the magnetic repulsion force can be calculated given the vertical displacement of the equipment using Equation **??**. This was implemented as a subsystem in Simulink, incorporating transformations necessary to determine magnet separation given horizontal displacement; repulsion force acting on a single bearing given magnet separation; and total reaction force given the force acting on a single bearing.

Ideally the damping coefficent for various masses would be determined empirically, however, using the theoretical approach outlined in Section 4.6.1 the damping coefficient was found for a range friction coefficients μ. Typically for these materials $0.4 < \mu < 0.6$. Substituting the limits and nominal mass into Equation **??**, the damping coefficient was bounded as follows

$$159 \text{ N s m}^{-1} < c < 238 \text{ N s m}^{-1}$$

Figure 15 on the next page displays the top level model architecture. The topology is typical of spring mass damper systems as detailed in [7].

The constant stiffness was replaced with the subsystem used to find the magnetic repulsive force. *SimIn* provided an interface to the MATLAB workspace which was used to input the driving force. Similarly, *SimOut* exported the system output to the workspace; the acceleration of the equipment as a time series. Using the inverse of Equation 1 on page 14, the model output was converted into voltages so its spectra could be compared to the accelerometer output measured for the undamped amplifier.

Figure 16 on the next page shows the effect of running the simulation with the damping coefficient at each extreme.

In both graphs, the fundamental frequency can no longer be observed; a peak at around 30 Hz is apparent however its low power and frequency that differs from 50 Hz indicate that this is just noise. The noise is effectively

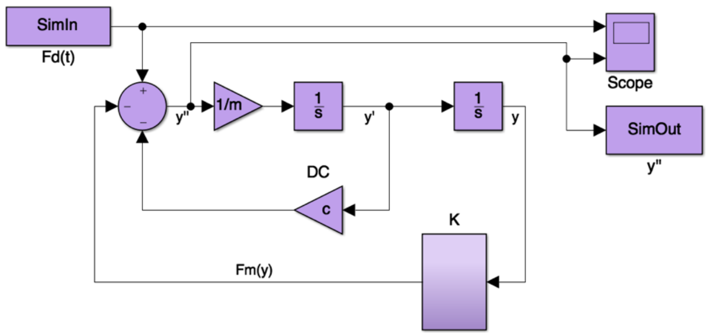


Figure 15: Compared Frequency Responses with and without Squash Balls

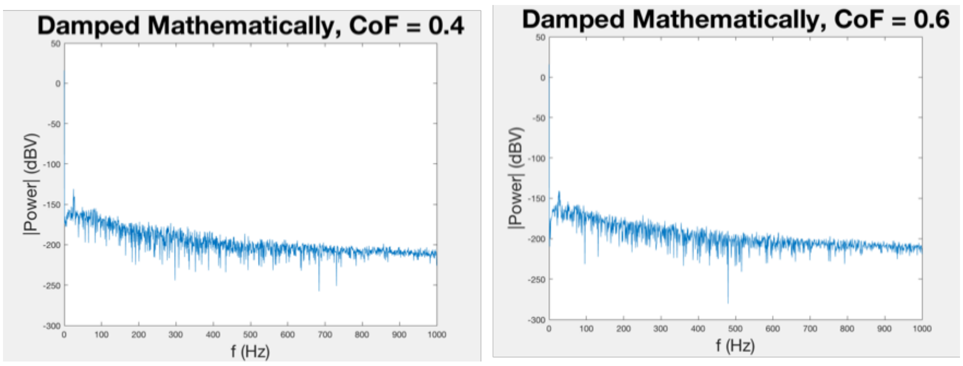


Figure 16: Power Output for Coefficient of Friction Extremes

identical in both cases and hence it can be concluded that the difference in damping coefficient across its range negligible effect. For further analysis, the median coefficient of friction will be used—0.5—this returns a damping coefficient of 198 $\mathrm{N\,s\,m^{-1}}$.

The undamped is displayed with the simulation output damped spectrum for comparison between the input and output of the dynamic model in Figure 17.

The red curve shows the simulation output; the fundamental frequency and its harmonics are indistinguishable from the noise. The entire signal has been reduced to below the noise floor of around -140 dBV. The noise has been almost completely attenuated.

The success of the footers is likely to be less significant than that of the model. The primary reason for this discrepancy is the value of damping coefficient used. Ideally, this would be found empirically; a series of prototypes would be produced and a similar experiment carried out to find the true damping coefficient.

If the footers have a fraction of the successs they have been predicted to

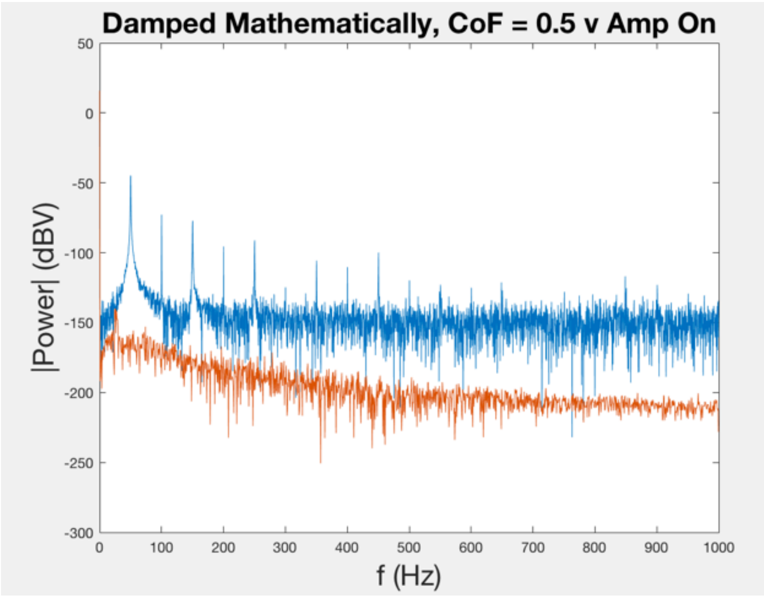


Figure 17: Comparison of Undamped and Damped Response

have, there will be a significant improvement in sound quality.

5 Detail Design

6 Design for Manufacture

DFM defines the design of a product so as to best optimise its quality whilst minimising its cost to manufacture[8]. Factors affecting this cost include the number of off-the-shelf and machinable parts, the set-up time of required machinery, the material type, dimensional tolerances as well as secondary processes. Generally, a compromise is reached between the functional quality of a product and the cost of manufacture however, considering the current extortionate pricing of similar existing products, certain design choices have taken precedence over their implications in a manufacturing context, for the example the rails within the complex central piece are undesirable to manufacture(see drawing COR080-0003) yet optimal for the specified mechanism.

6.1 Bill of Materials

Table 1: Bill of Materials

Item No.	Part No.	Description	Qty.
1	COR080-0003	Isofonics core piece	1
2	TOP080-0004	Isofonics top piece	1
3	10MMTUNGSTENBALLS[1]	⌀10 mm tungsten carbide ball	6
4	F669-N45SH-10[2]	⌀10 x 1.5 mm neodymium button magnet	12
5	PLD010-1004	Isofonics preloading back-stop	1
6	RET080-0003	Isofonics retainer	1
7	M4X20-CSK-ST[3]	M4 x 20 mm T20 A2 c'sunk screw with partial thread	6
8	PLD080-0003	Isofonics preloading crown	1
9	M4X20-CSK-H[4]	M4 x 20 mm H2.5 A2 c'sunk screw with partial thread	1
10	RET080-1002	Isofonics retainer bottom	1
11	USR080-0001	Isofonics removable spike	1
12	USR080-1002	Isofonics removable base	1

[1] http://www.vxb.com/10mm-Tungsten-Carbide-Bearing-Ball-0-3937-inch-Dia-p/10mmtungstenballs.htm
[2] http://www.first4magnets.com/circular-disc-rod-magnets-c34/10mm-dia-x-1-5mm-thick-n45sh-neodymium-magnet-1-1kg-pull-p3633
[3] http://www.westfieldfasteners.co.uk/A2_ScrewBolt_PinTXCsk_M4.html
[4] http://www.westfieldfasteners.co.uk/A2_ScrewBolt_SHCsk_M4.html

6.2 Off-The-Shelf Parts

Due to the complex geometry required from our design, only few components may be bought in, namely (per mount); six Ø10mm tungsten carbide ball bearings, six ⌀10 x 1.5 mm neodymium magnets, six M4 by 20 mm AISI A2 steel countersunk Torx security screws and one M4 by 20 mm countersunk hex socket with partial thread. These components are readily available excluding the tungsten carbide ball bearings which must be sourced from a specialist supplier. Table 1 details potential sources for the aforementioned parts and the quantity required per mount.

6.3 Primary Processes

All parts are to be machined using a 3 axis CNC mill excluding the central piece (see drawing COR080-0003), which also requires wire erosion. Wire erosion is considerably more expensive but allows for the more intricate designs required by this piece. Forging was considered as a method of manufacture but was dismissed for the following reasons; the large expenditure involved in machinery, dies, tools and personnel are only justifiable for large scale production and the die accuracy required to forge the most complex pieces is likely unachievable[9]. In the interest of reducing manufacturing cost, for which the complex geometry of the core piece is largely responsible, it was aproposed during discussion with Sylatech Ltd®that the piece be machined in separate parts that are later joined to avoid the machining of reverse chamfers as shown in Figure 18.

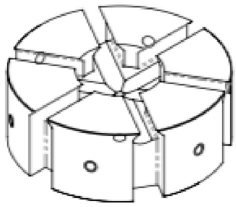


Figure 18: Core

Cost is substantially driven by time; time to remove material in the machining process as well as set-up time of the machine itself[10]. Conveniently, our design is highly symmetrical meaning time and money is saved through lacking the need of a complex orienting mechanism given that the part's orientation prior to machining is irrelevant.

Finally, the volume of production is of great importance; with too small a batch size, set up costs and jig production costs become impractical and with too large a batch, storage costs pose a problem for a product of which the market response is not easily predictable. Considering the high cost and exclusive appeal of this product, low volume batch production in the order of 100 is appropriate so as to eliminate any unnecessary storage costs and allow adequate response to market needs. Specifically a volume of 300 pieces has been defined because it is divisible by three, the mounts are expected to be sold in groups of three, and during correspondence with Sylatech Ltd®it was discussed that any larger batch volume would pose an infeasible machining time; namely roughly two weeks.

6.4 Secondary Processes

A range of finishes best suited to individual parts have been chosen based on its functionality and location. A relatively cheap 2B (basic smooth) finish is to be applied to the stainless steel core so as to prohibit unnecessary expenditure considering the part shall not be seen by the customer whilst ensuring a path of minimal friction exists for the moving ball bearings. Additionally, the standard aluminium mill finish is rough and so all internal aluminium preloading pieces are to be brushed in the direction the piece travels so as to limit the amount of wear. Furthermore, the external retainer is to be brushed in concentric circles to give an attractive finish and hide any scratches. A 2J finish is to be applied to all other external stainless steel pieces as it is cheaper to produce than polishing and is practical in that it is resistant to scratches whilst being aesthetically pleasing[11].

All parts are to be machined at fine linear and angular tolerances of +/-0.1mm and +/-1° respectively to ensure the overall quality of the product. However there exists opportunity for optimisation within this section; in the interest of reducing cost, it was concluded that the rails contained in the core piece are the only parts to be machined to a high tolerance since they are to fit plush to the bearings; all other parts may be machined to a coarser, and therefore cheaper, tolerance.

6.5 Manufacture Costs

6.5.1 Optimisation

Sylatech Ltd®, a machining company situated in the North East of England, was contacted regarding the cost to manufacture the product and proved very informative with regard to the feasibility of our product. Initially a cost of roughly £500 per piece (for a batch volume of 300) was quoted which, allowing for appropriate margins detailed in section 9.3, dictates a retail price of £1650. The most expensive existing product of similar concept is priced at £849.99 and, although our original product features merit a higher retail price, a difference of roughly £800 is too large when introducing a new product to the market.

The following suggestions were made to reduce the quoted figure;

- The rails of the core piece were to be modified such that the radius of the bearing race was continued until tangential with the wall. Simply, the right-angles depicted in Figure 19 were to be removed. This revision was quoted to save roughly £100 per mount.
- The 0.3mm dimension of the base as shown in Figure 20 was to be extended to 2mm; the larger dimension is easier to grip and therefore would require less delicate and costly machine handling. This change was quoted to save roughly £10 per base.
- The retainer was to be machined using a 4 axis CNC machine which is considerably faster because fewer operations are required. This change was quoted to save roughly £20 per piece.

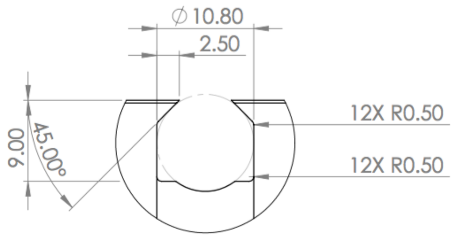


Figure 19: Bearing Race Cross Sectional Drawing

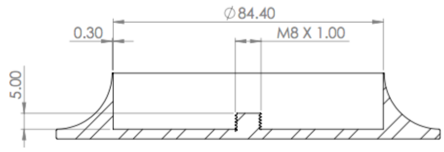


Figure 20: Base Cross Sectional Drawing

6.5.2 Costing

Table 2: Costing

Item No.	Part No.	Qty.	£/Piece	£ Total
1	COR080-0003	1	97.00	97.00
2	PLD080-0003	1	35.00	35.00
3	PLD010-1004	6	2.00	12.00
4	F669-N45SH-10 (first4magnets)	12	0.30	3.60
5	10MMTUNGSTENBALLS (VXB Bearings)	6	20.30	121.80
6	TOP080-0004A	1	32.50	32.50
7	RET080-0003	1	25.00	25.00
8	M4X20-CSK-ST (westfieldfasteners)	6	0.09	0.54
9	M4X20-CSK-H (westfieldfasteners)	1	0.04	0.04
10	USR080-0002	1	14.30	14.30
11	RET080-1002	1	12.00	12.00
12	USR080-1002	1	20.00	20.00
				373.78[1]

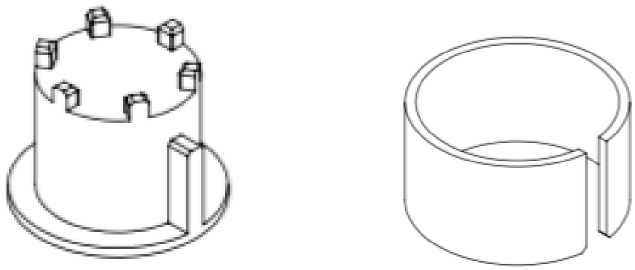
[1] Sylatech Ltd, Kirkdale Road, Kirkbymoorside, North Yorkshire, YO62 6PX, Quote courtesy of Mr R McGill

Table 2 details the final cost of off-the-shelf parts and machining per mount, totalling 393.78 per mount.

7 Design for Assembly

7.0.1 Jig Design

Due to the small scale of our product, hand assembly is required. Considering the complexity of the design and physical impracticality of overcoming the repulsive magnetic forces during its assembly, a jig has been designed, as shown in drawings JIG080-0001 and JIG080-1001, with full accompanying assembly instructions (see Isofonics Assembly Instructions ISO080-INS).



(a) Assembly Jig Base (b) Assembly Jig Sleeve

Figure 21: Assembly Jig Components

7.1 Assembly Costs

Considering the assembly jig is to be re-used indefinitely, its cost is negligible within the context of the product's assembly. Assembly costs are then only defined by the time taken to manually assemble the device and the cost of manual labour which can be estimated at 2 minutes and £8/hr respectively. This equates to a total of £0.27 per piece.

8 Design for Sustainability

Isofonics recognises the importance of considering the environmental implications of the manufacture as well as disposal of any product.

Isofonics footers are made to last a lifetime. This is achieved by using the highest quality materials possible; quality and endurance were more important than the price of the product considering the high budget of the design. This reduces the harm done to the environment by eliminating the need to manufacture replacement footers.Another method of ensuring minimal impact is to reduce the scrap, this is achieved by ensuring that the size of the raw material is a similar size to the final product. The main central section of the footer is produced using a CNC machining process as described previously and so the un-refined steel will be cut to within a small tolerance of the final product. The scrap that is inevitably produced will be recycled; this includes both stainless steel and aluminium.

Stainless steel contains valuable raw materials, especially chromium, nickel and molybdenum and hence recycling it is economically viable. Like many other metallic materials, stainless steel is recycled through a re-melting process, where the melted steel is re moulded for use. Similarly, aluminium is also recycled with a re-melting process and can be reprocessed and reformed endlessly without losing any of its quality.

Packaging often contributes to the waste generated by the product; to avoid this, the footers will be packaged in simplistic cardboard based materials which can be efficiently recycled rather than plastics where the recycling process yields less useful products.

9 Commercial Considerations

9.1 Brand Development and Competitor Analysis

With the intention of establishing the design in the market, a brand identity was essential. Through several brainstorming sessions, the team agreed on the name 'Isofonics '.

It was important to review the current market leaders, Stillpoints and Nordost, with their relevant products. Nordost primarily manufactures Hi-Fi audio cables but recently introduced additional products for resonance and power control. Namely, their 'SortKone'acts as a vibration drain with a mechanical diode effect to prevent external vibrations from travelling up through the cone. Similarly, 3 cones are recommended per device but they offer 3 types of cone, each made from different materials. The most expensive is made from titanium and utilises ceramic bearings, retailing at £349.99 per cone.**Nordostcost** Stillpoints offer an equivalent solution, their most expensive is their ?Ultra 6,? priced at £799.99 per mount and £849.99 with an accompanying base. **StillpointsCost**

9.2 Stillpoints Patent Check

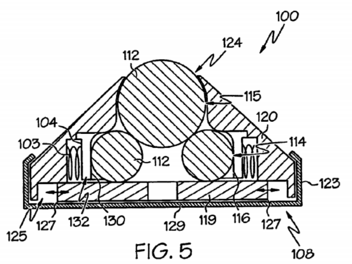
Stillpoints'patent for their 'Universal Vibration Damper'was thoroughly studied and the potential risks of infringement were identified. Their patent details the design of a 'device for the control of vibrations comprising a retainer resting on a base and a plurality of bearings disposed within the retainer.'It focuses mainly on the use of layered bearings and springs to diminish the signals through friction with three or more on the first layer and at least a substantially larger one on the second. Each bearing on the first layer is held by the base plate and side of the retainer and supports the bearing above. It also explores other options to replace the springs such as using opposing magnets: 'In some embodiments the first base member 119 and second base member 123 may have opposing magnetics fields.'[12] However, Isofonics? product is not an embodiment of the mechanism described in the first claim of the patent. A cone with a retaining flange is used rather than the layers of spherical bearings.

9.2.1 Parallels with Stillpoints

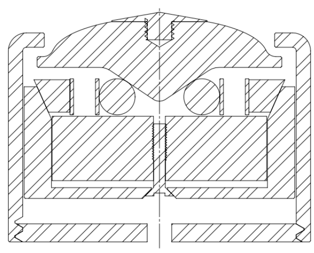
The following details the main claim made by Stillpoints and highlights areas of concern:

‘A device for the control of vibrations comprising: *a retainer*, the retainer constructed and arranged to rest upon a base, at least a *portion of the base defining a substantially flat surface*; and *a plurality of bearings disposed within the retainer, the bearings arranged in a first layer and a second layer*, the second layer disposed on the first layer, the first layer comprising three or more bearings, and the second layer comprising at least one bearing, each bearing in the first layer constrained on its bottom by only the substantially flat surface of the base, on its side by the retainer, the bearings in the first layer supporting the at least one bearing in the second layer, the retainer defining a surface which is in substantially tangential contact with the bearings in the first layer’[12].

It is clear that the main claim focuses on the use of layers of bearings, in this way, our design is fundamentally different. Since our design differs from that outlined in the main claim, all further claims based on the first are irrelevant. An element of originality of our design lies in the use of magnets which is not mentioned at all within the official claims section of the patent.



(a) Stillpoints Mechanism Diagram



(b) Group 25 Mechanism Diagram

Figure 22: Stillpoints Mechanism Comparison

To verify that the mounts did not bear a resemblance to Stillpoints'invention, Robin Bartle, a patent lawyer from Bartle Read, a company based in Liverpool which specialise in intellectual property was contacted who was able to confirm that 'Because the mount does not feature a second layer of bearings as shown on the cross-section, there are no blatant overlaps between the two products and Stillpoints would not have sufficient grounds to sue'(see Figure 22). Additionally, Mr Bartle noted that this is a US patent and is not filed with the European Patent Office and so only stands to limit sales within the US.

9.2.2 Points of Originality

- Use of *opposing magnetic fields* for lateral damping as opposed to springs
- *Cone with retaining flange* replacing layers of bearings
- *Pre-loading mechanism* in which a screw in tension moves a fixed magnet decreasing its distance from the bearing magnet thus increasing the strength of the system—adjustable for a variety of masses.

9.3 Marginal Costs

With reference to Table 2, and accounting for assembly costs as discussed in section 7.1, the main contained mechanism, optional base and spike are to cost roughly £340 , £20 and £14.30 respectively to produce. With respect to production costs, margins of 100% for product distribution, 100% for retail and 30% for profit are to be met which consequently dictate the following retail prices. The main device shall retail for £1,149.00, the optional base for £66.00 and the removable spike for £49.00.

10 Discussion

10.1 Specification Fulfilment

10.1.1 User Requirement Specification Tree

With reference to the URS tree (see Figure 1), the final proposal incorporated a cone with a retaining flange which held the magnets in place whose separation was altered in order to accommodate a range of masses. With reference to the URS tree (see Figure 1), the four degrees of freedom were thus defined by all translational motion and free rotation. As described earlier in the report, the magnet behind the bearing is moved while the other is fixed and the preloading mechanism allowed the user to adjust the position of the fixed magnet. Namely, the crown element moved the back stops for the fixed magnet inward as a single preloading screw was tightened.

The requirement for a rigid load path was integral to the performance of the damping mechanism. The interaction between the magnets and the spherical bearings in conjunction with the hardness of their material clearly diminished the excess mechanical vibrations generated by microphonic effects as demonstrated by the graphs produced by the dynamic model. However, the data yielded during the experiment which was then inputted to the simulation, did not totally reflect the execution of the design. Employing the squash balls as a method of dampening the noise produced by the amplifier did validate the concept but this could be supported further by utilising a prototype and later determining the damping coefficient for different loads empirically. If these results demonstrated a fraction of the model?s success, then the mount can still be confidently viewed as a success.

The Hi-Fi equipment must only rest on 3 footers to prevent further resonances and heating in the arrangement. If another was used, it would either prove redundant or induce further noise. The cone provided a single point of contact which in turn meant that 3 footers were needed to support one piece of equipment. Partnered with the large base plate, this guaranteed that the mounts would not slip and the system was amply stable. For this reasoning, the footers would be sold in sets of three.

A 2-D finite element analysis software was used to model the magnetic flux distribution within the mount. Although it was concluded that there was no additional interference or damage to the Hi-Fi system, ideally a 3-D finite element analysis software would better determine whether the separation distance or the grade of magnet needed to be amended.

REDO for SPECIFIC LINKS TO TREE. Split into 3 main aims - - Eliminate microphonic effects - show proof of damping, single point of contacts means coupled to equipment and hard materials ensures rigid paths, use of circular bearings has allowed for all four degrees of freedom.

-Integrate with existing equipment- comment on aesthetics (material finish), 3 points of contact are sufficient for stability, define range of masses it can support. Improvements- no proof of not blemishing existing equipment

-Budget- basically have no budget, talk about current market price.

10.2 Further Developments

There is great potential for further work and like Nordost, a family of products adapted to support heavier devices such as loudspeakers could also be developed by changing the material for the spherical bearings or by varying the magnetic field strength to yield a new damping factor and related coefficient. A companion phone application which enables the user to adjust the preload for a desired condition could also replace the preloading graphs provided by the parametric model. Machining could be improved by optimising tolerancing for the tracks and will thus increase profit margins and set the product aside from the mentioned competitors.

11 Conclusion

12 References

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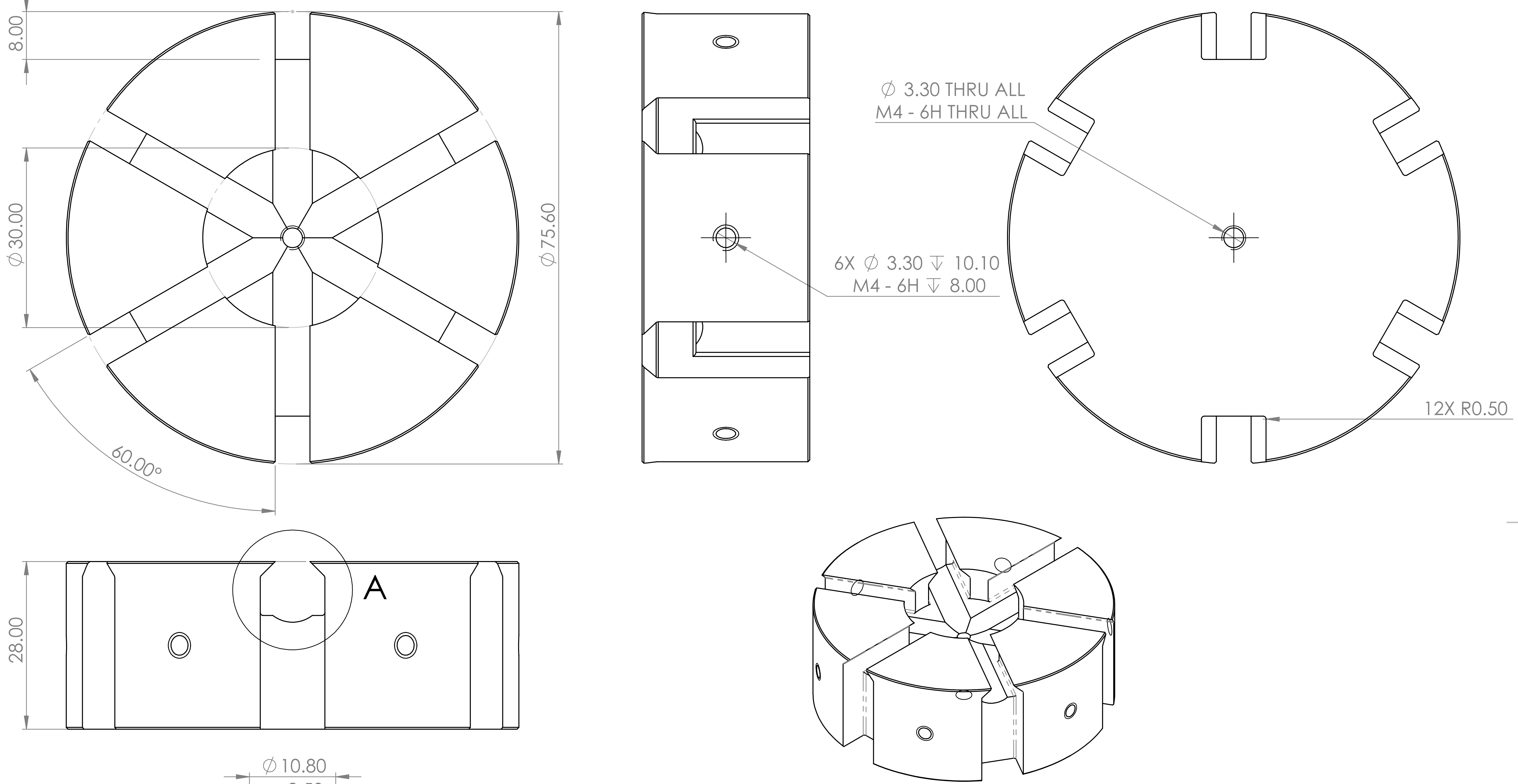
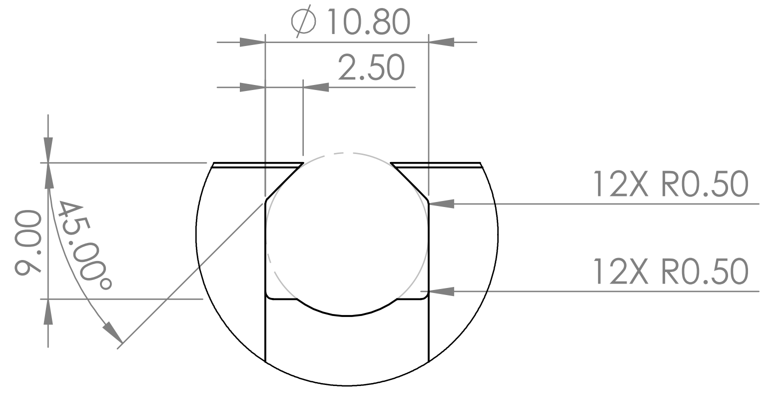
[11] M. H. David Cochrane, *Guide to stainless steel finishes*, T. Edition, Ed. Euro Inox, 2005.

[12] W. et al, "Universal vibration damper", Stillpoints, Tech. Rep. USA Patent US 6,655,668 B1, 2003.

A Project Plan and Members' Contributions





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DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE: Isofonics core

MATERIAL: AISI 304 Stainless Steel

DWG NO. COR080-0003

A3

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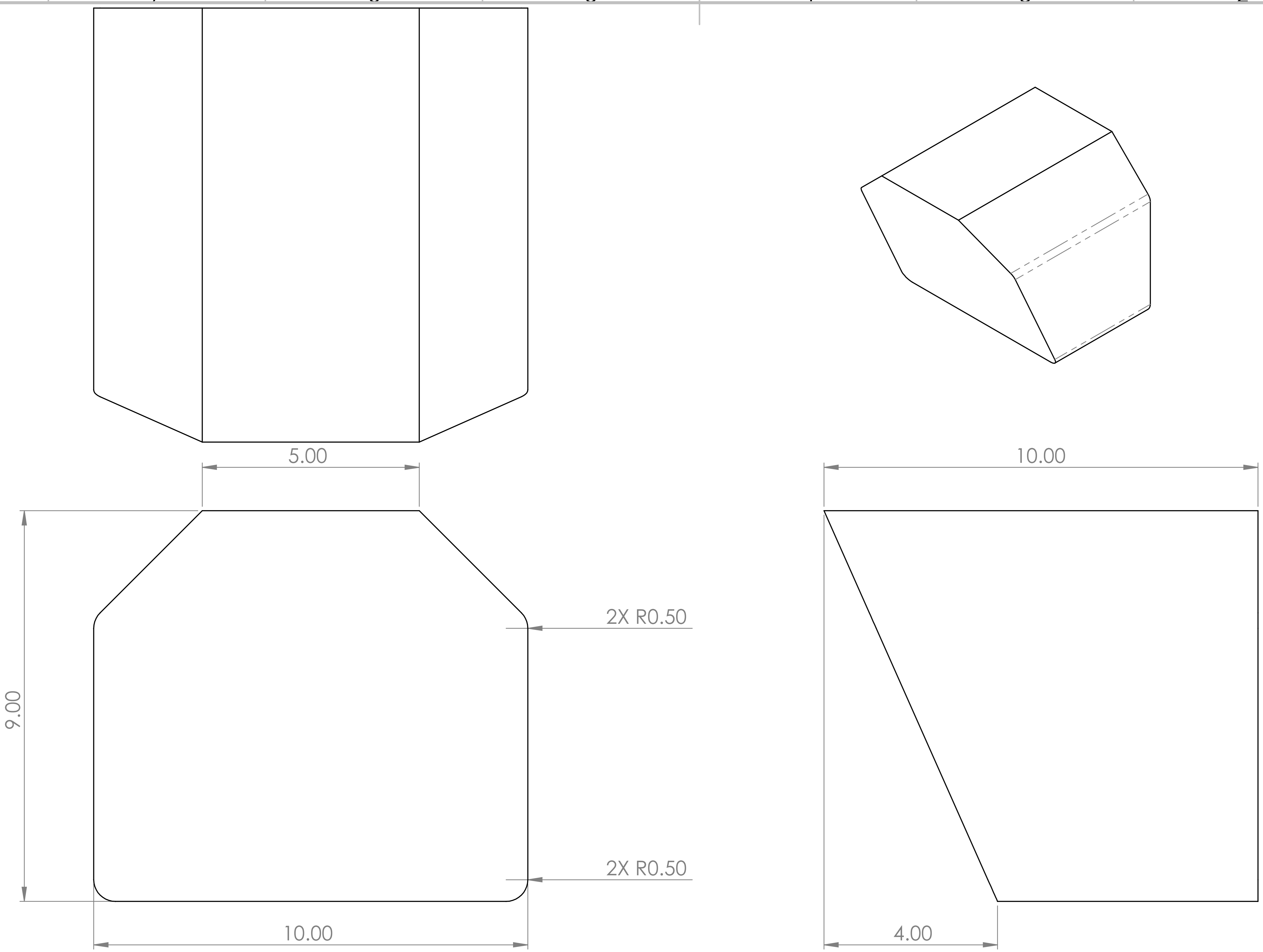

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DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE:	Isofonics preloading backstop
MATERIAL:	6061-T6 Aluminium
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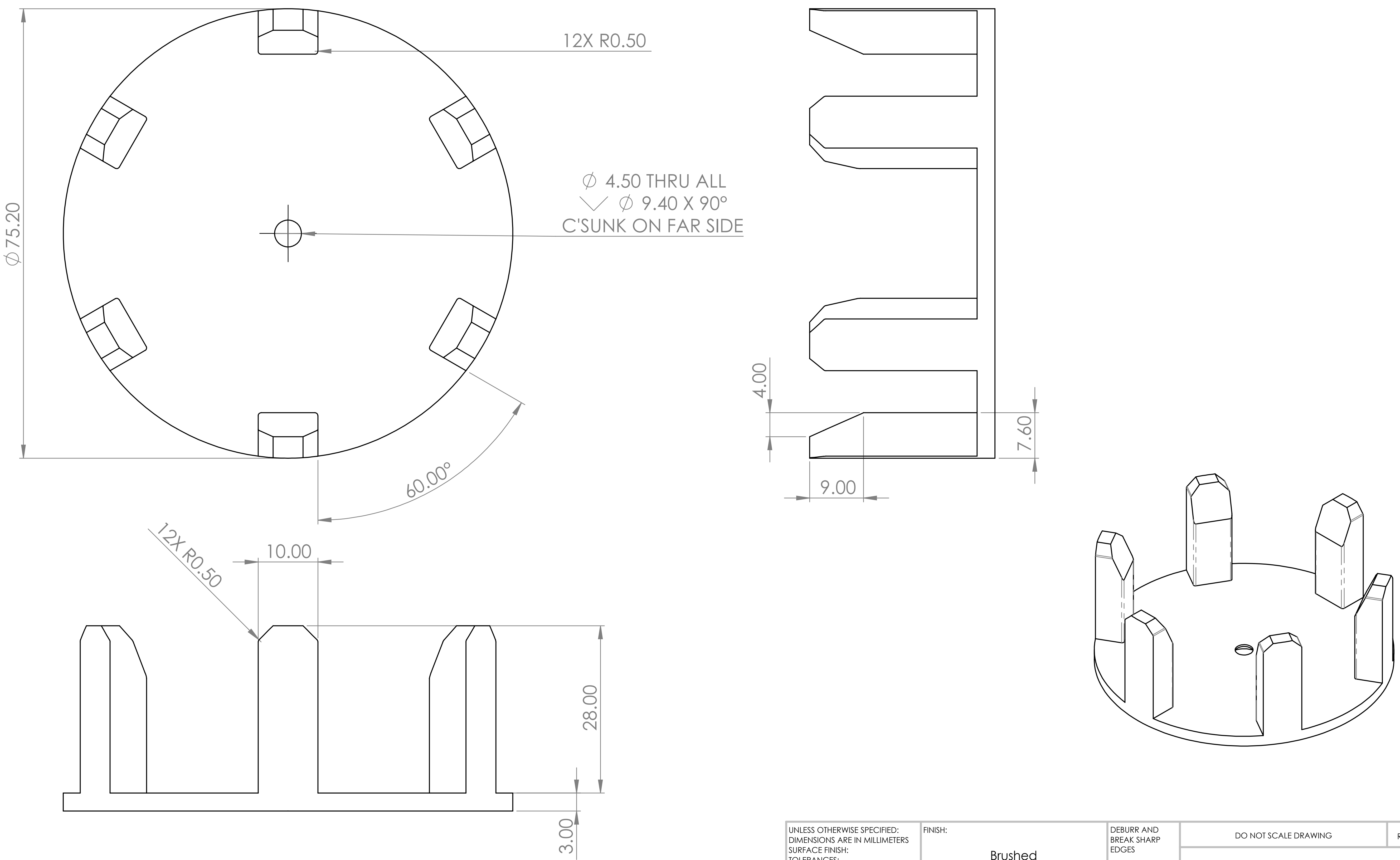

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ANGULAR: ± 1 deg

FINISH: Brushed

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

MATERIAL: 6061-T6 Aluminium

WEIGHT:

TITLE: Isofonics preloading crown

DWG NO. PLD080-0003

A3

SCALE:3:2

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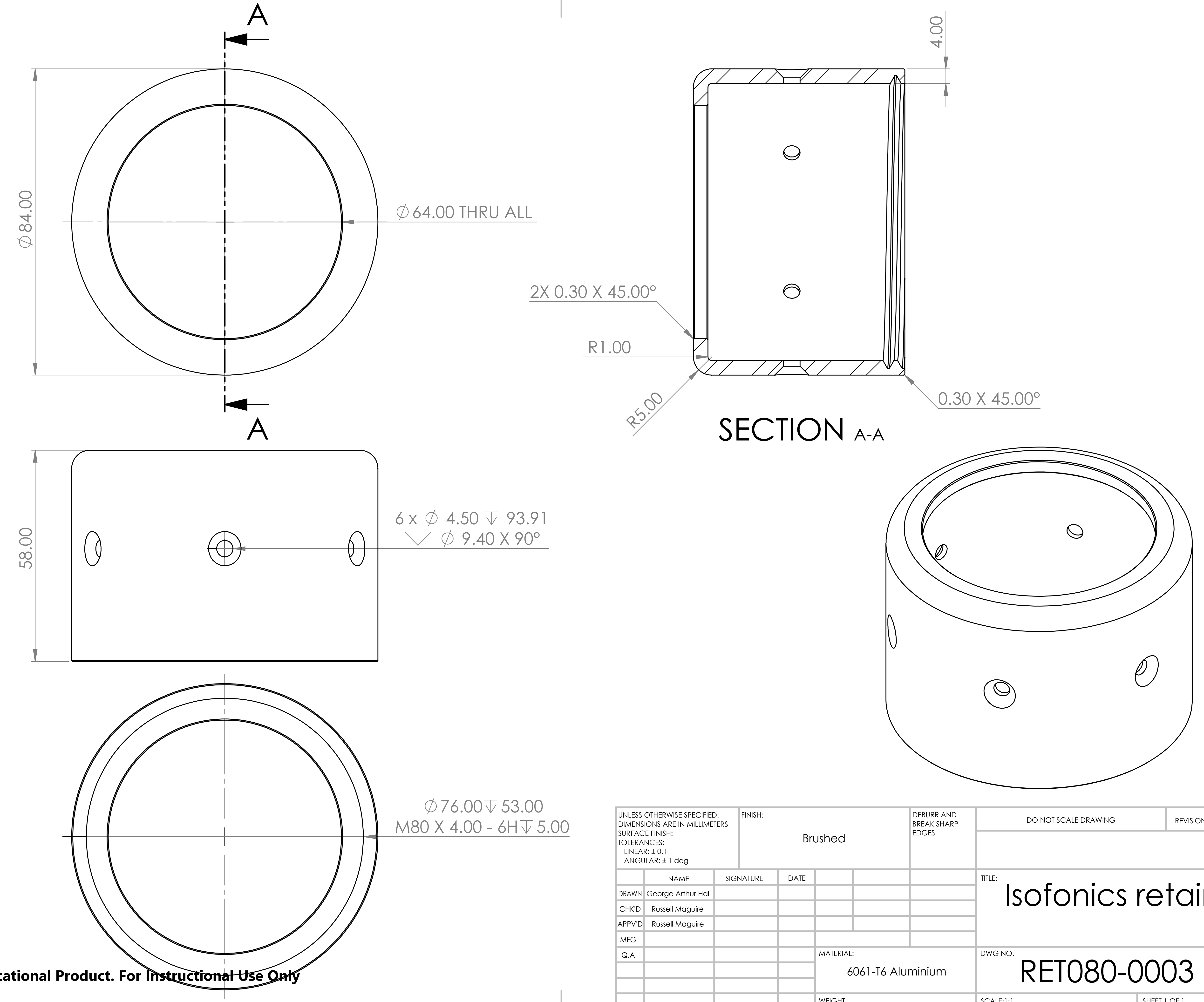

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DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE: Isofonics retainer

MATERIAL: 6061-T6 Aluminium

DWG NO. RET080-0003 — A3

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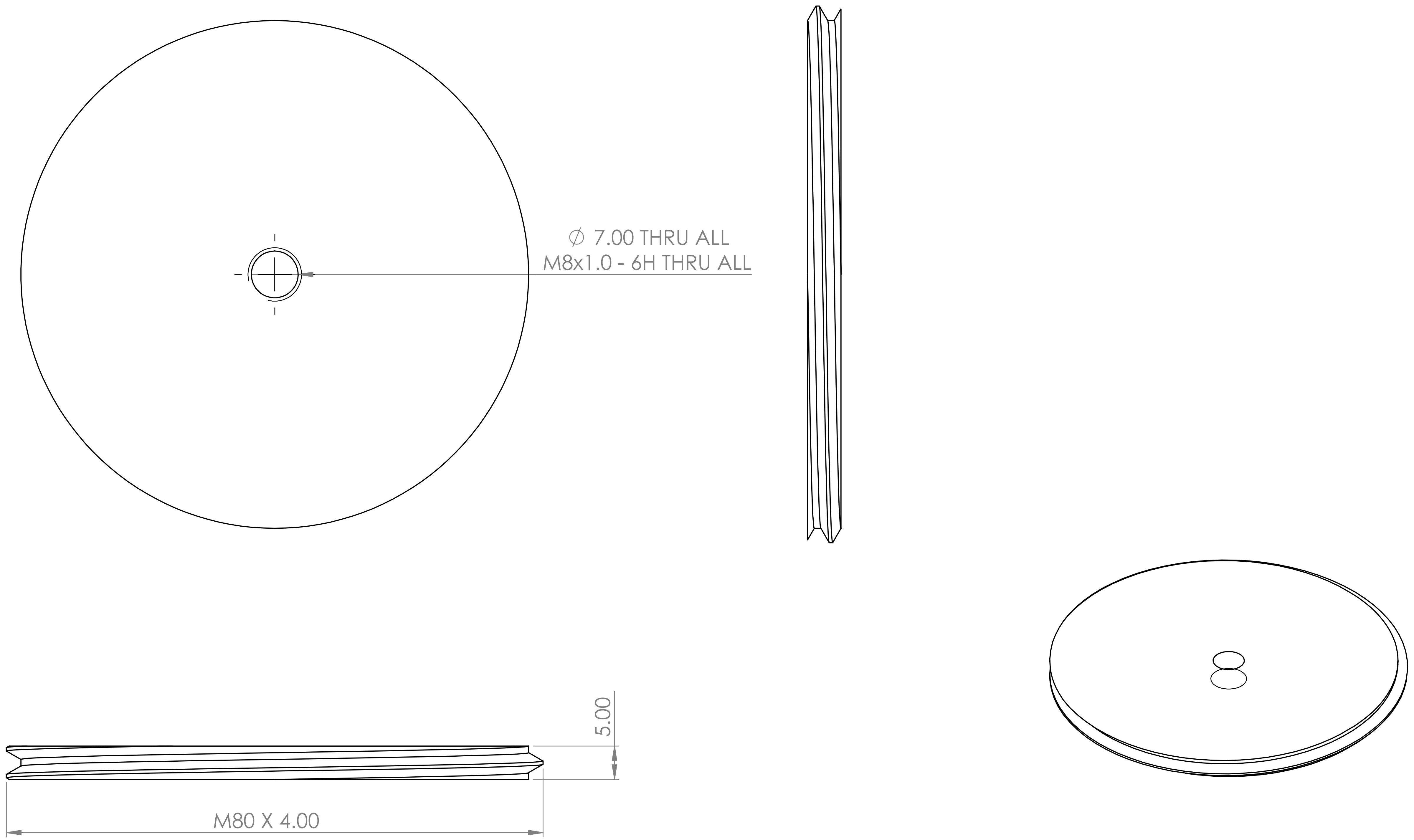

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DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE:

Isofonics retainer bottom

MATERIAL: 6061-T6 Aluminium

DWG NO. RET080-1002

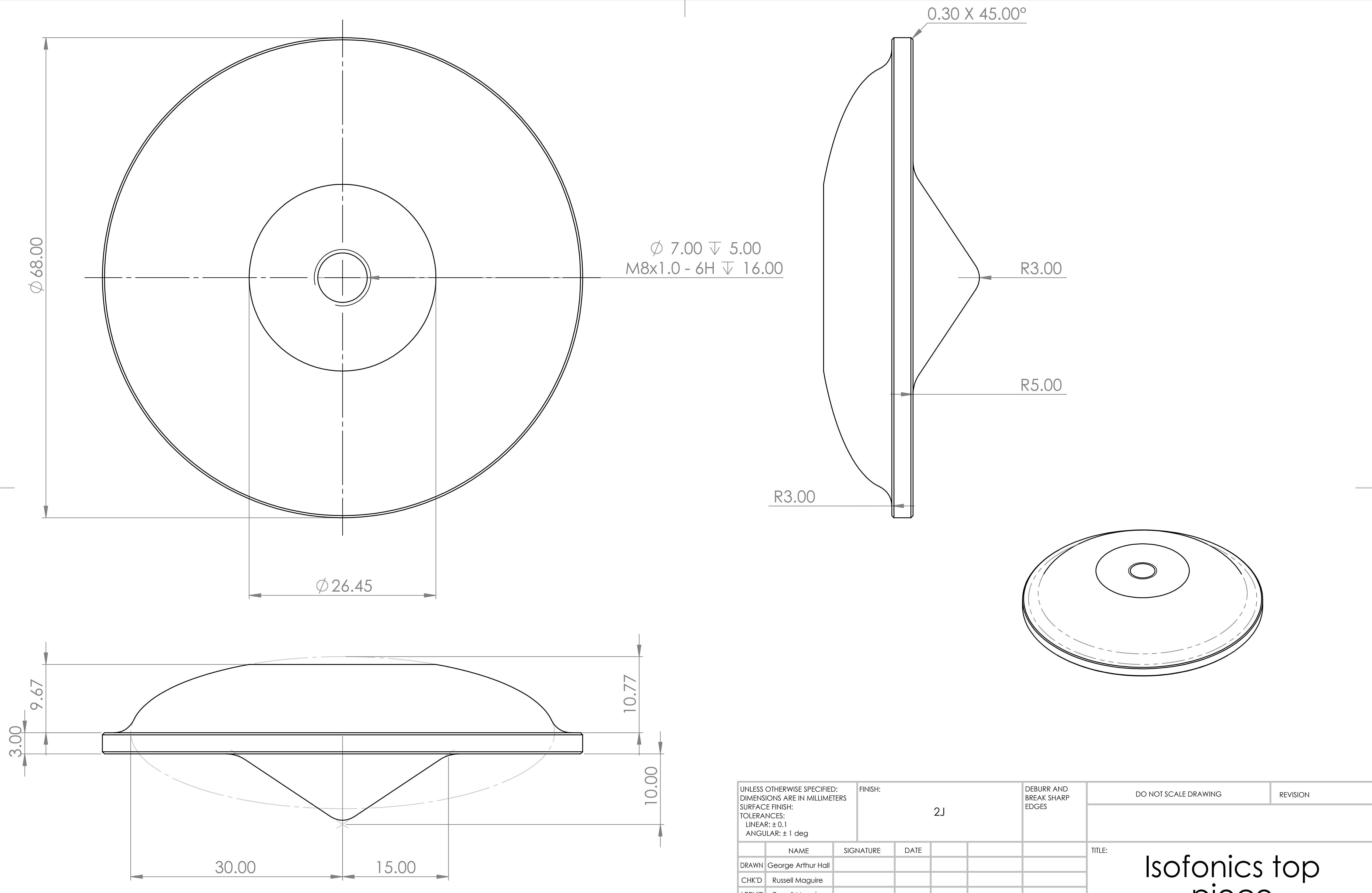
A3

WEIGHT:

SCALE:3:2

SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only



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	NAME	SIGNATURE	DATE
DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE: Isofonics top piece

MATERIAL: AISI 304 Stainless Steel

DWG NO. TOP080-0004

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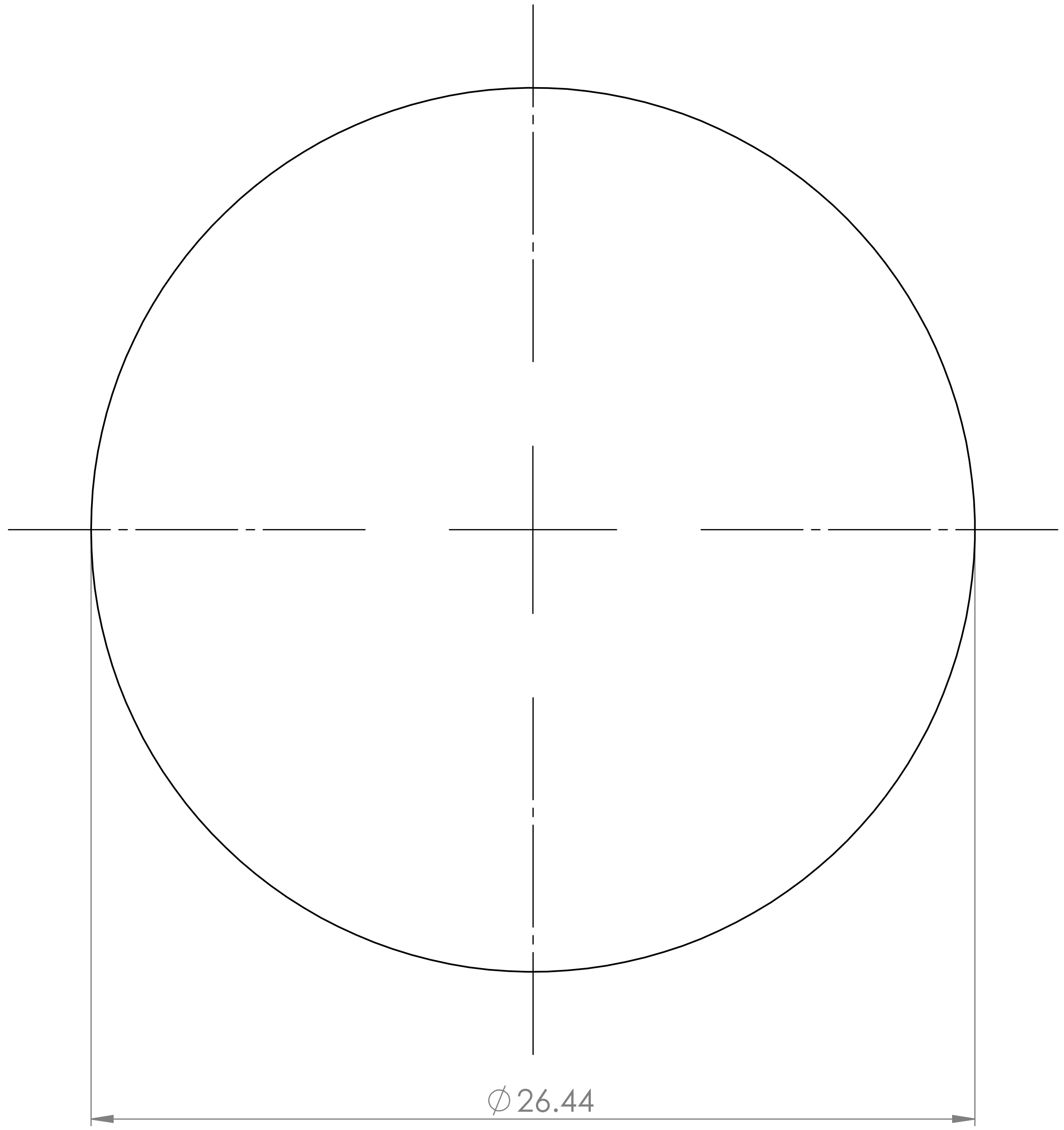

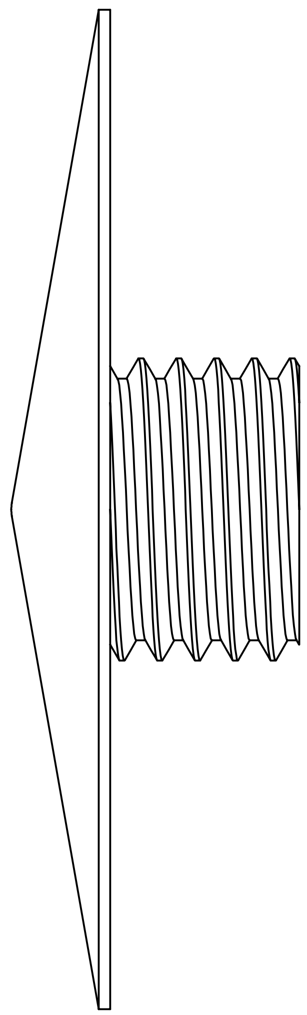

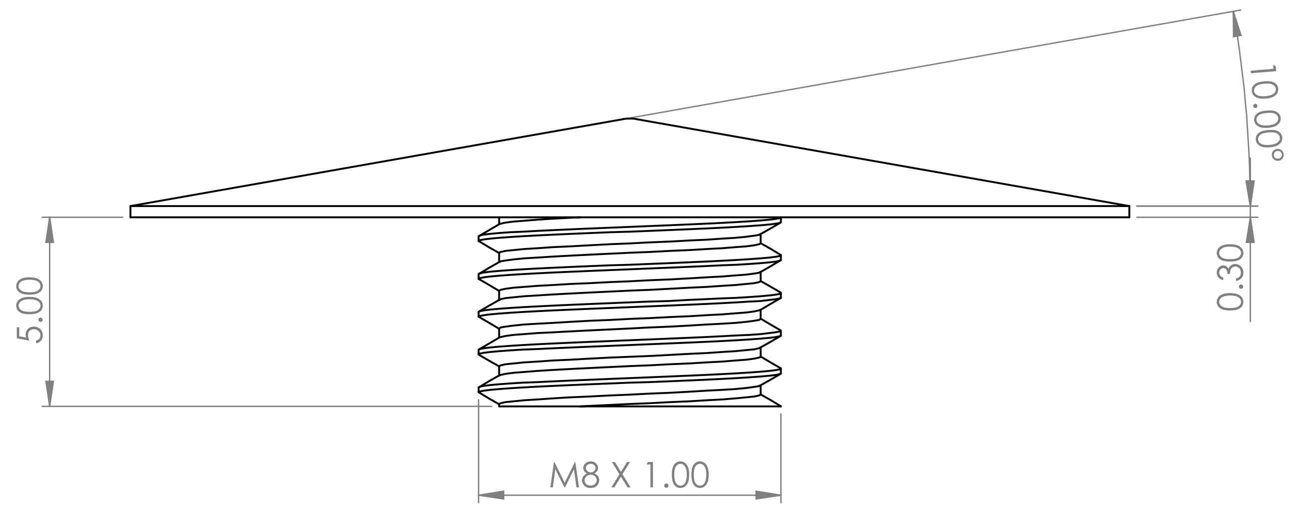

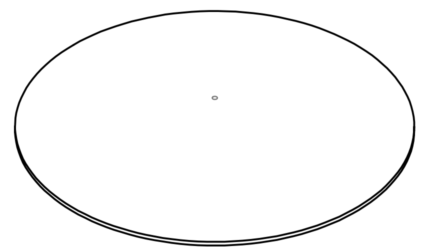

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	NAME	SIGNATURE	DATE
DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE:

Isofonics removable spike

MATERIAL: AISI 304 Stainless Steel

DWG NO. USR080-0002

A3

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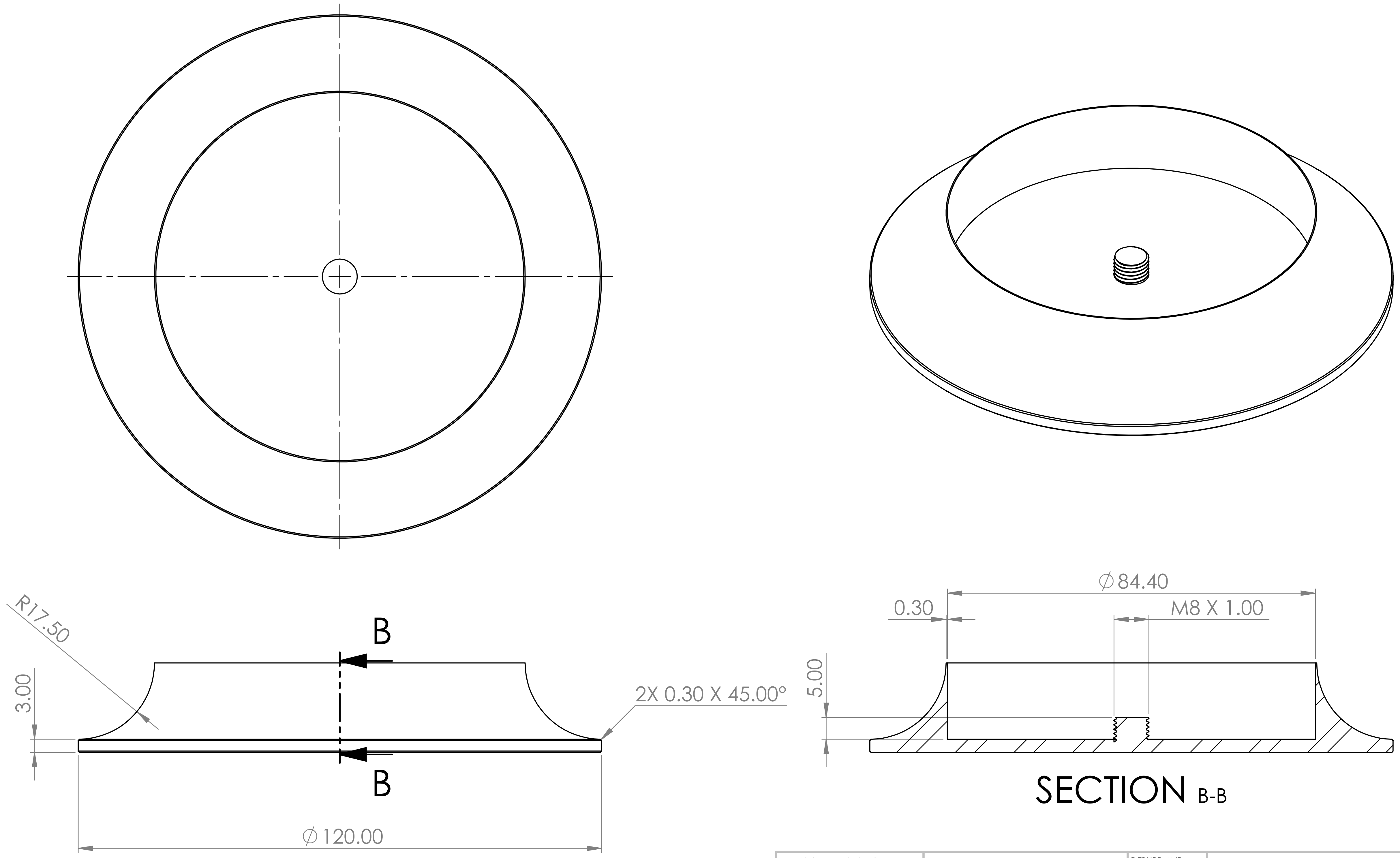

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FINISH: Brushed

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN	George Arthur Hall		
CHK'D	Russell Maguire		
APPV'D	Russell Maguire		
MFG			
Q.A			

TITLE: Isofonics removable base

MATERIAL: 6061-T6 Aluminium

DWG NO. USR080-1002

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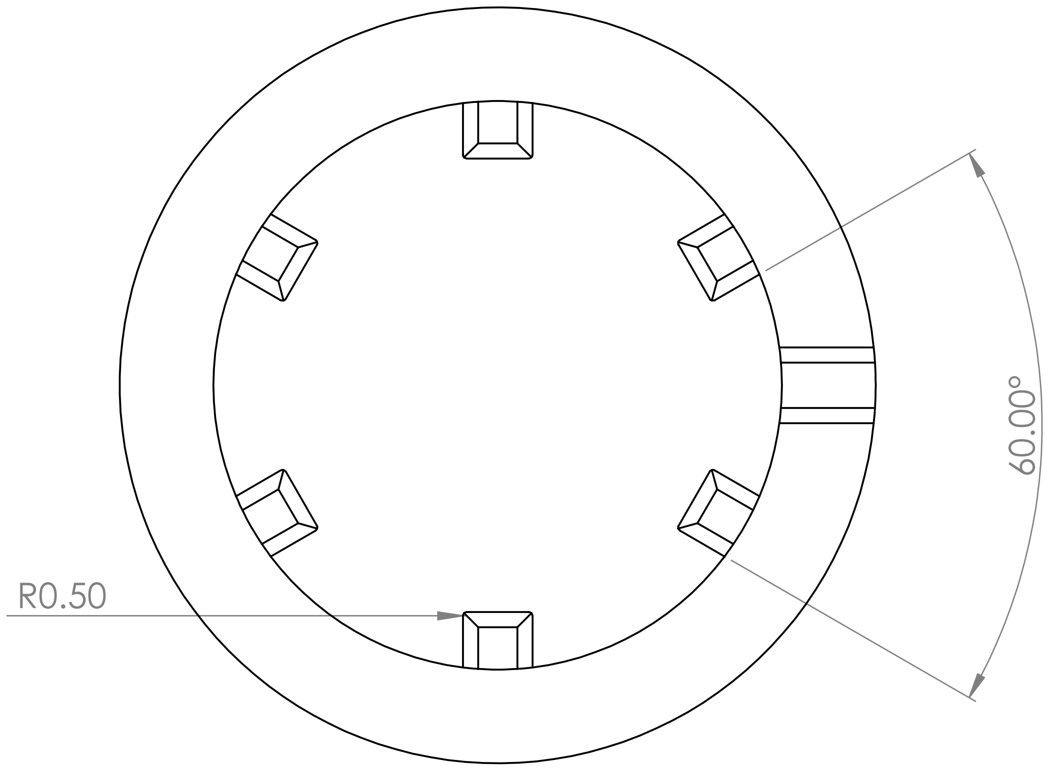

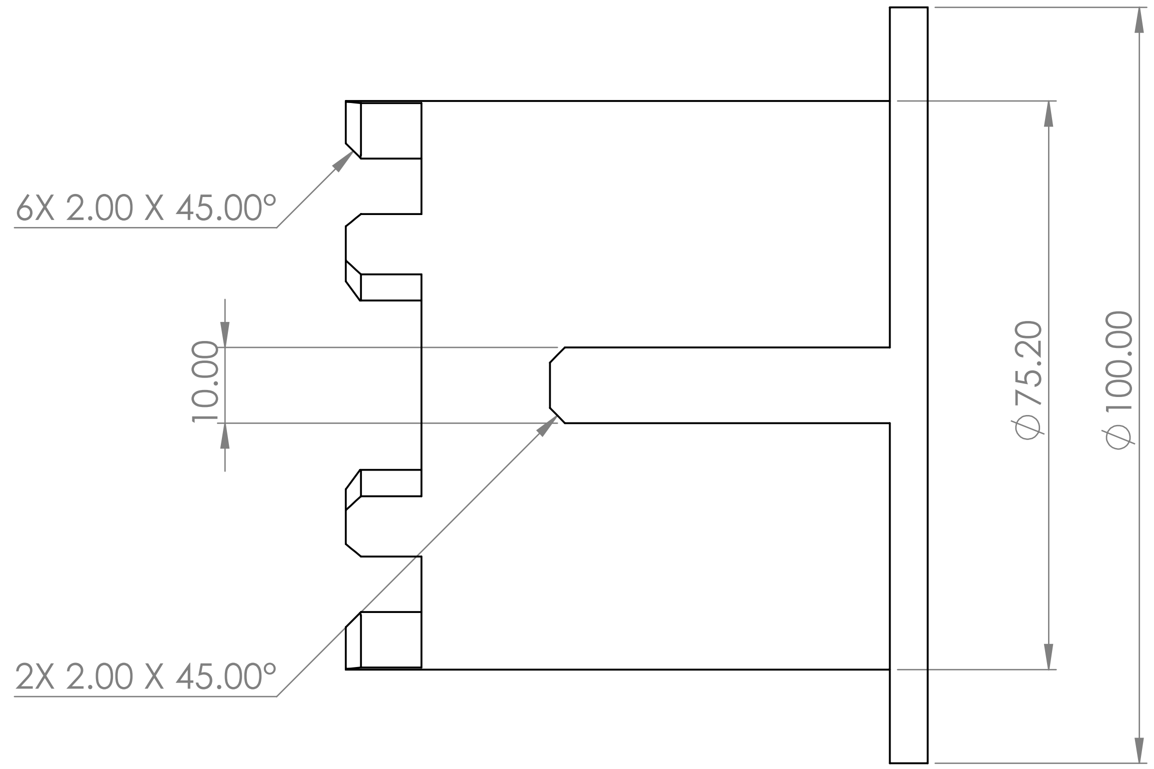

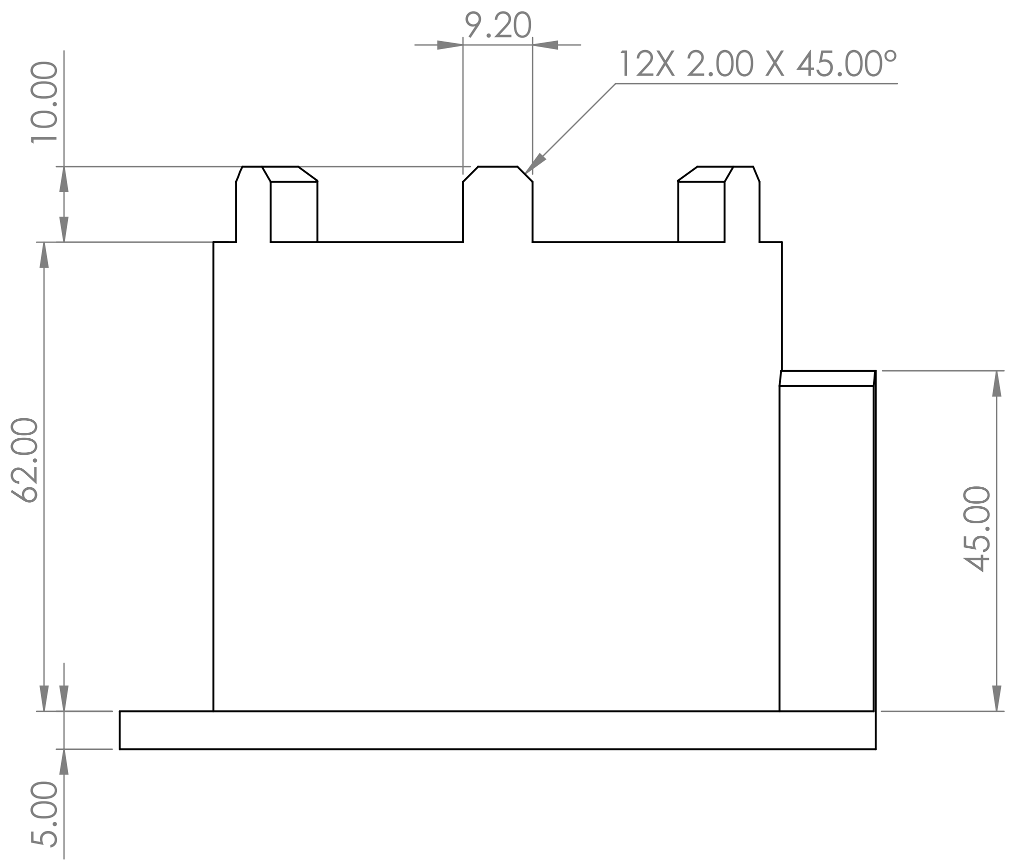

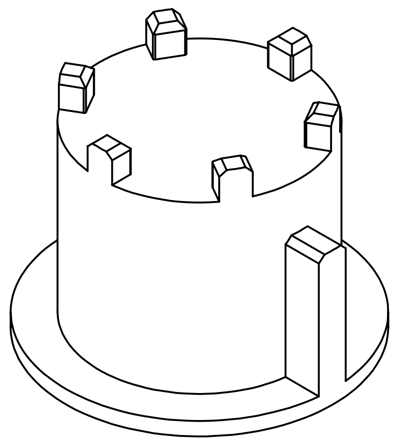

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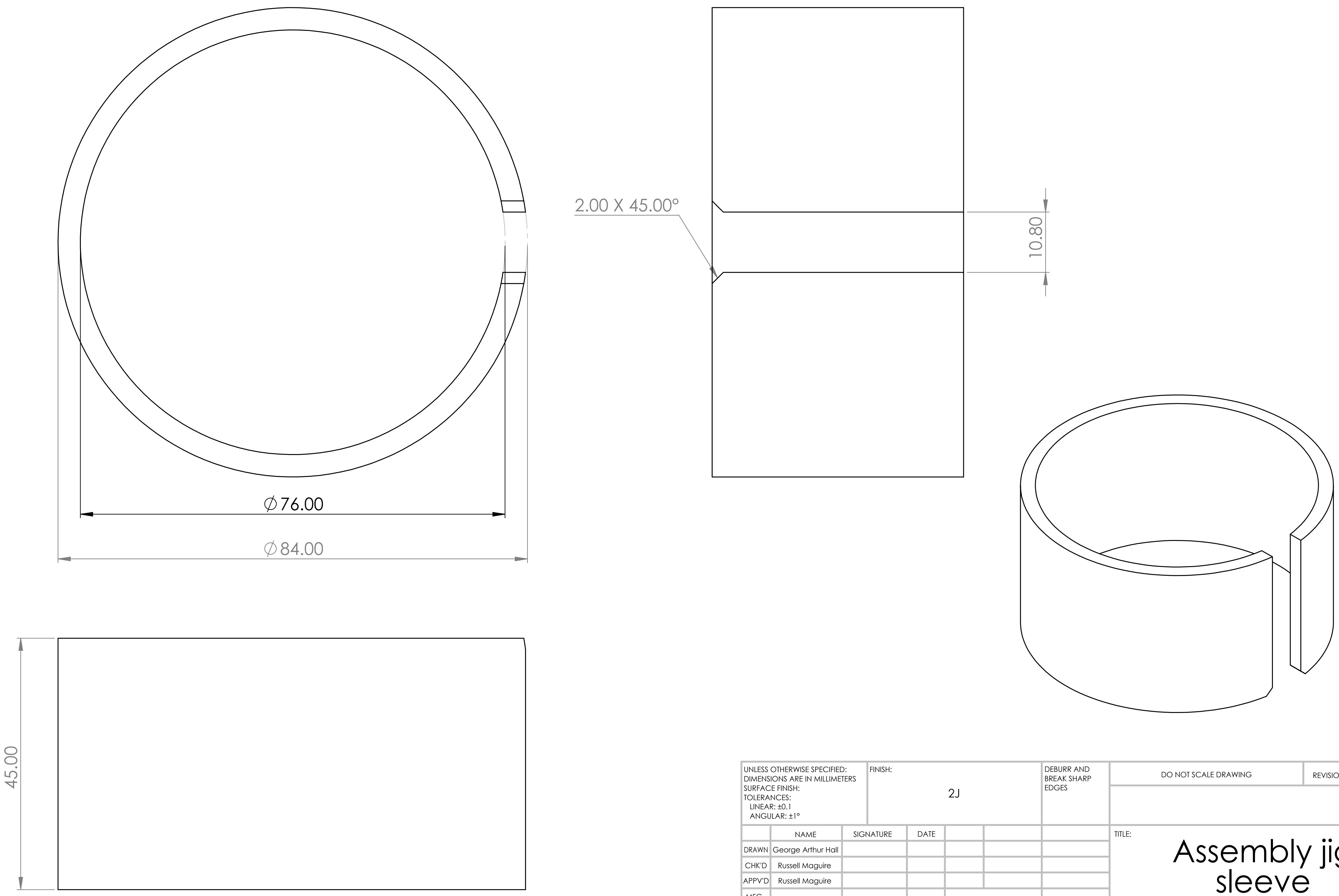
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ±0.1 ANGULAR: ±1°	FINISH: 2J	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE	TITLE: Assembly jig base
DRAWN	George Arthur Hall			
CHK'D	Russell Maguire			
APPV'D	Russell Maguire			
MFG				
Q.A			MATERIAL: AISI 304 Stainless Steel	DWG NO. JIG080-0001 A3
			WEIGHT:	SCALE:1:1 SHEET 1 OF 1



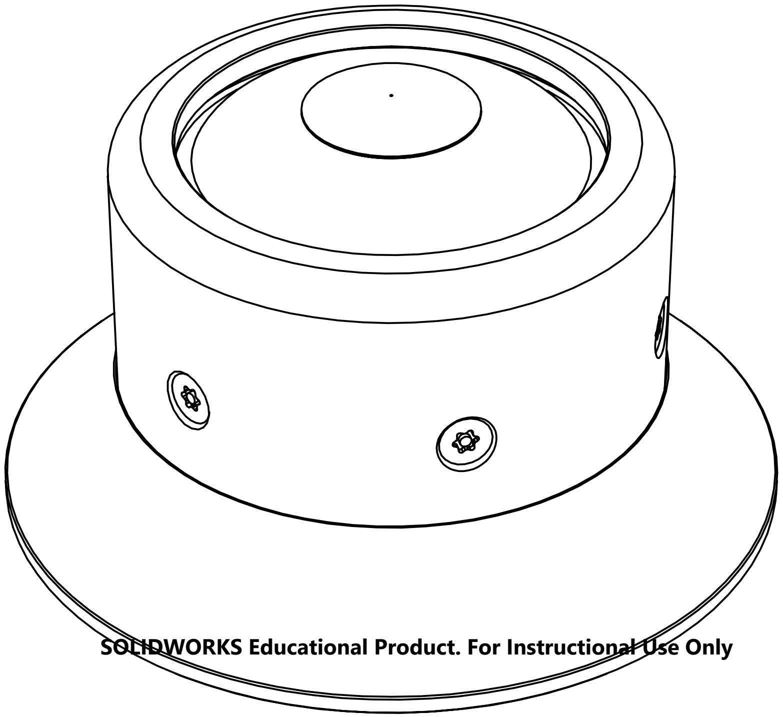
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ±0.1 ANGULAR: ±1°	FINISH: 2J	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION	
	NAME	SIGNATURE	DATE	TITLE: Assembly jig sleeve	
DRAWN	George Arthur Hall				
CHK'D	Russell Maguire				
APPV'D	Russell Maguire				
MFG					
Q.A			MATERIAL: AISI 304 Stainless Steel	DWG NO. JIG080-1001	A3
			WEIGHT:	SCALE:3:2	SHEET 1 OF 1

ISOFONICS

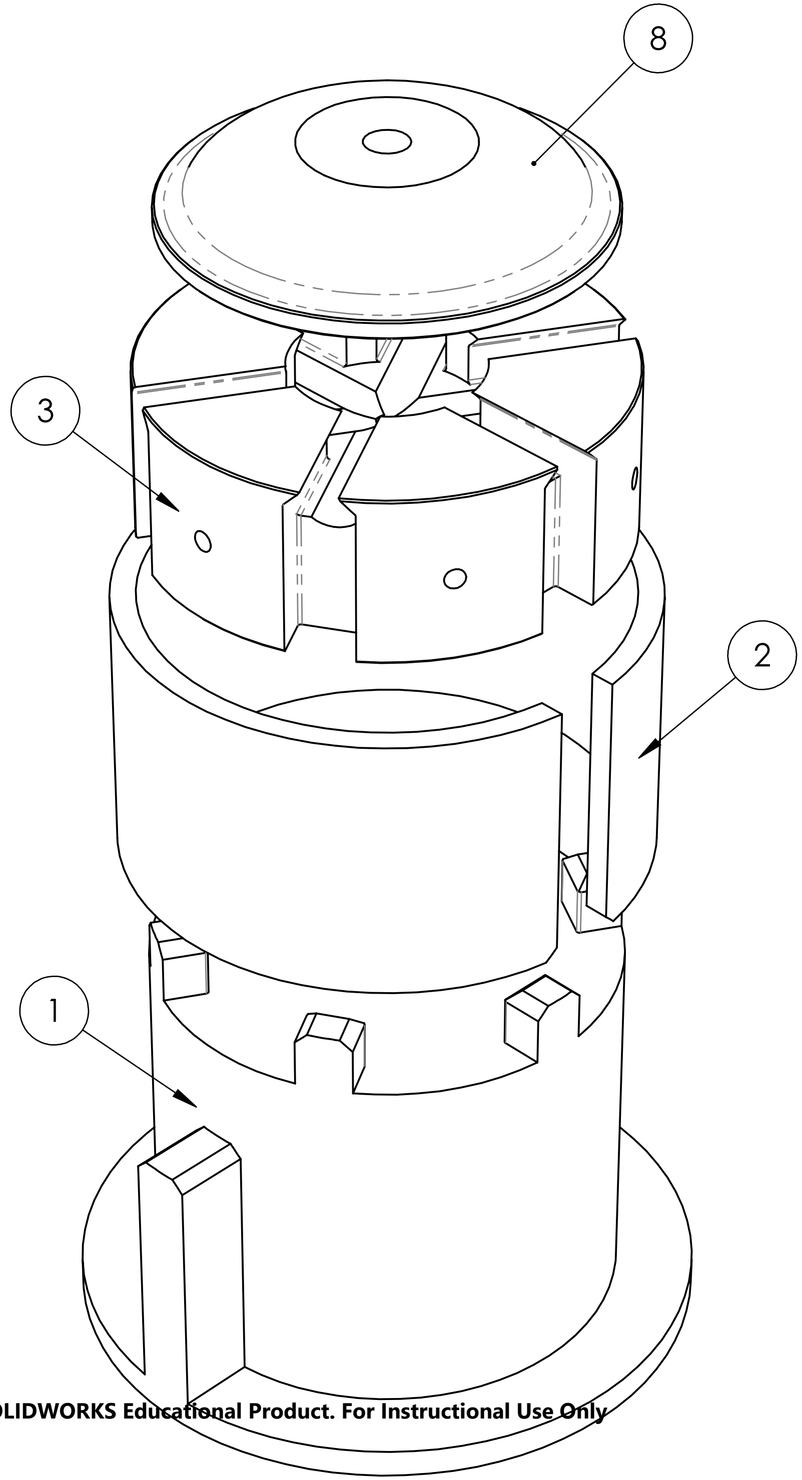
Assembly Instructions

ISO080-INS

R. S. Maguire

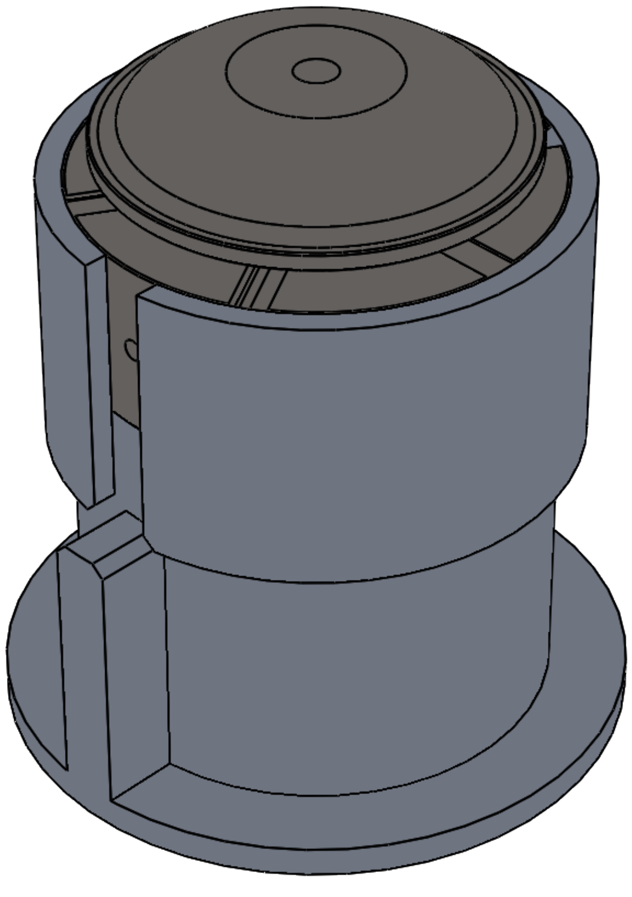


ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	JIG080-0001	Assembly jig base	1
2	JIG080-1001	Assembly jig sleeve	1
3	COR080-0003	Isofonics core	1
4	PLD080-0003	Isofonics preloading crown	1
5	PLD010-1004	Isofonics preloading backstop	6
6	F669-N45SH-10 (first4magnets)	10 x 1.5mm neodymium magnet	12
7	10MMTUNGSTENBALLS (VXB Bearings)	10mm tungsten carbide ball	6
8	TOP080-0004A	Isofonics top piece	1
9	RET080-0003	Isofonics retainer	1
10	M4X20-CSK-ST (westfieldfasteners)	Partially threaded M4 x 20mm c'sunk security screw	6
11	M4X20-CSK-H (westfieldfasteners)	Partially threaded M4 x 20mm c'sunk hex socket screw	1
12	USR080-0002	Isofonics removable spike	1
13	RET080-1002	Isofonics retainer bottom	1
14	USR080-1002	Isofonics removable base	1



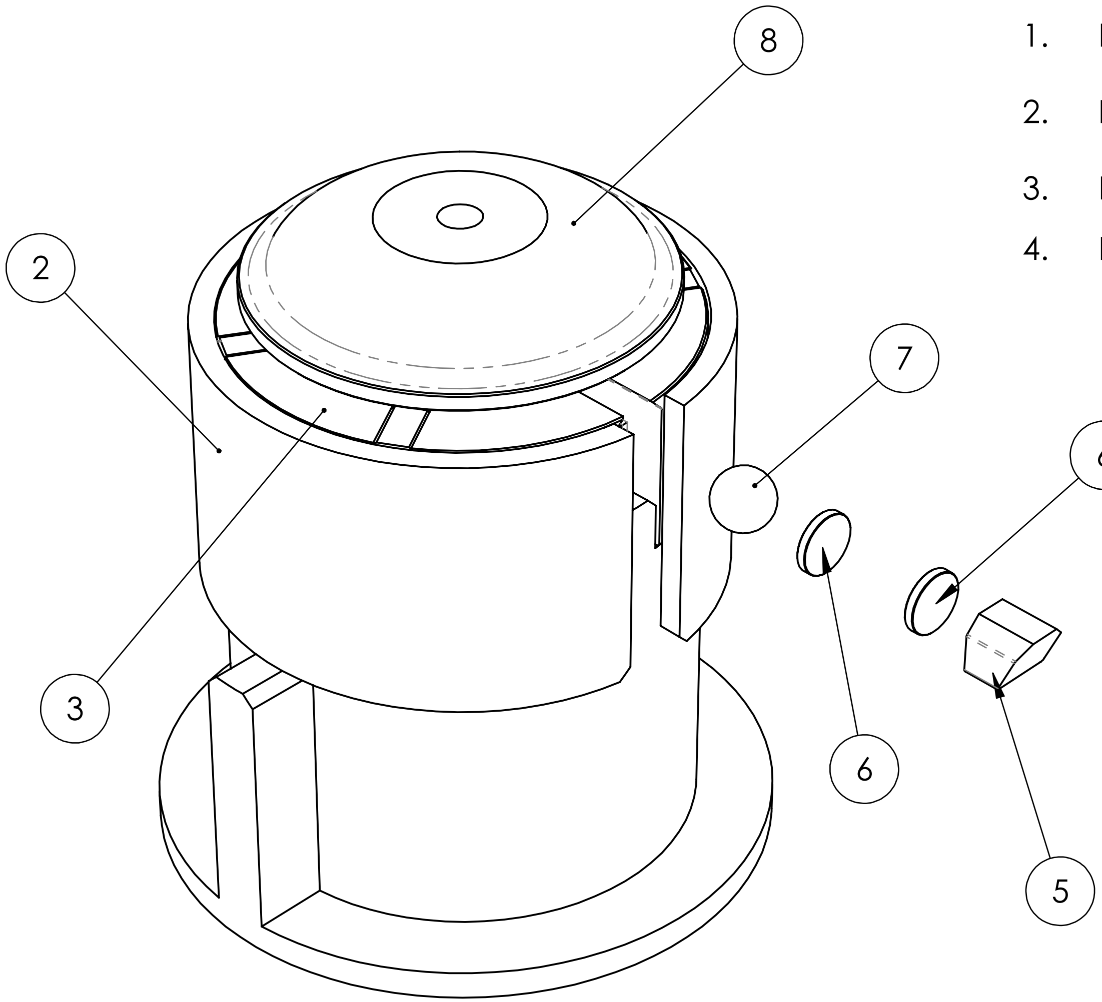
Step 1

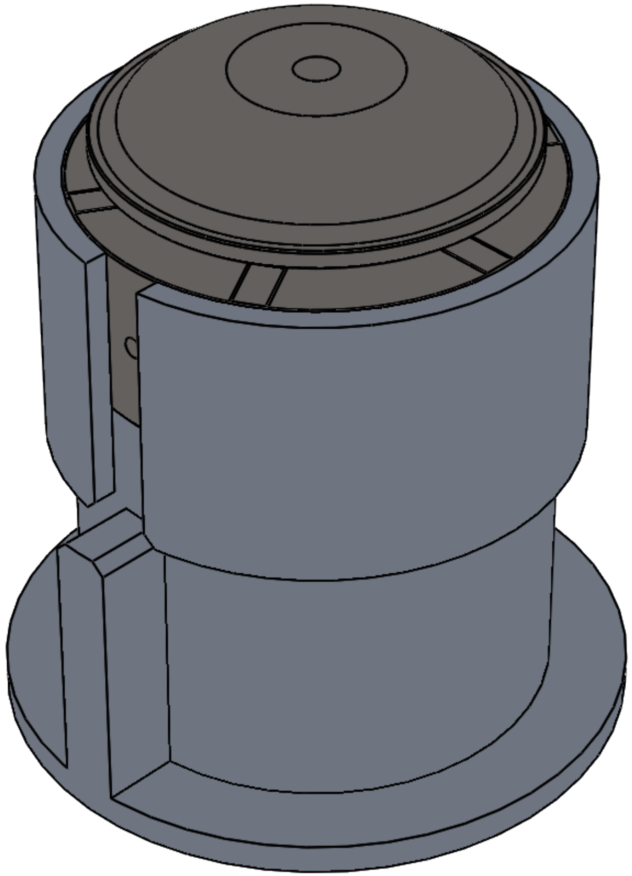
1. Slot 3 into 1
2. Slide 2 over 1
3. Place 8 ontop of 3



Step 2

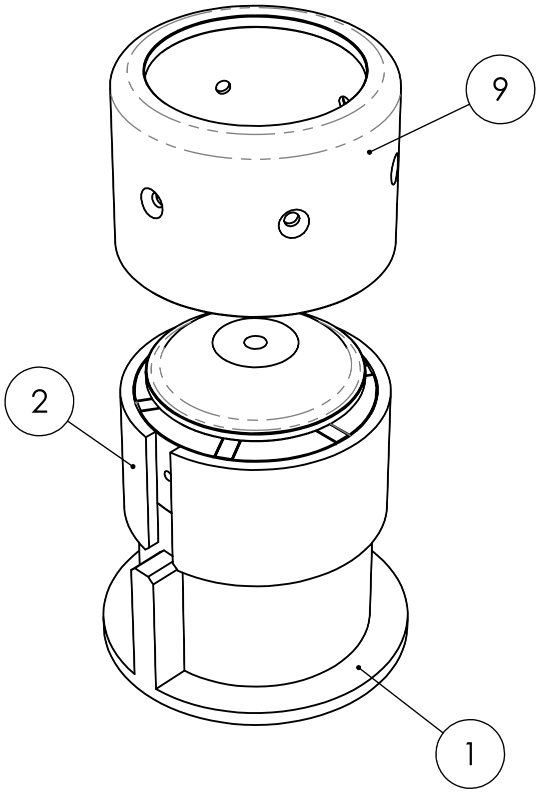
1. Rotate 2 until parallel with track in 3
2. Holding firmly onto 8, slide parts onto track
3. Rotate 2 so parts are contained
4. Repeat for all six tracks

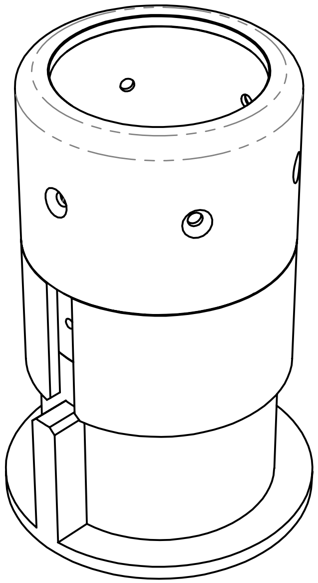


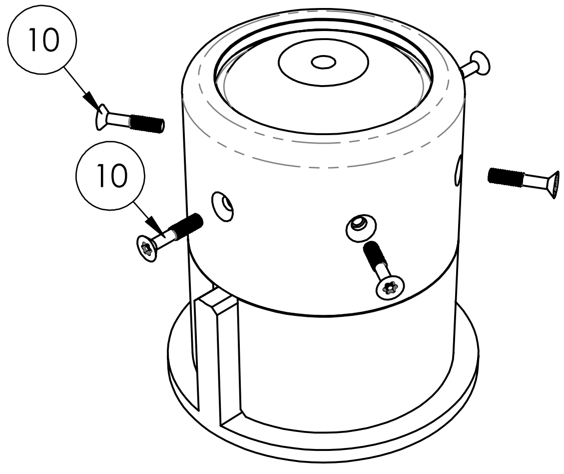


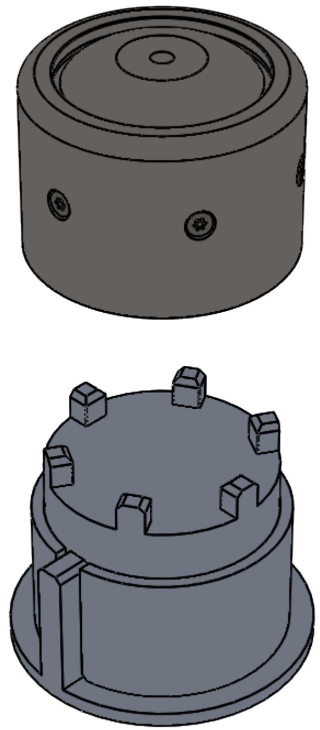
Step 3

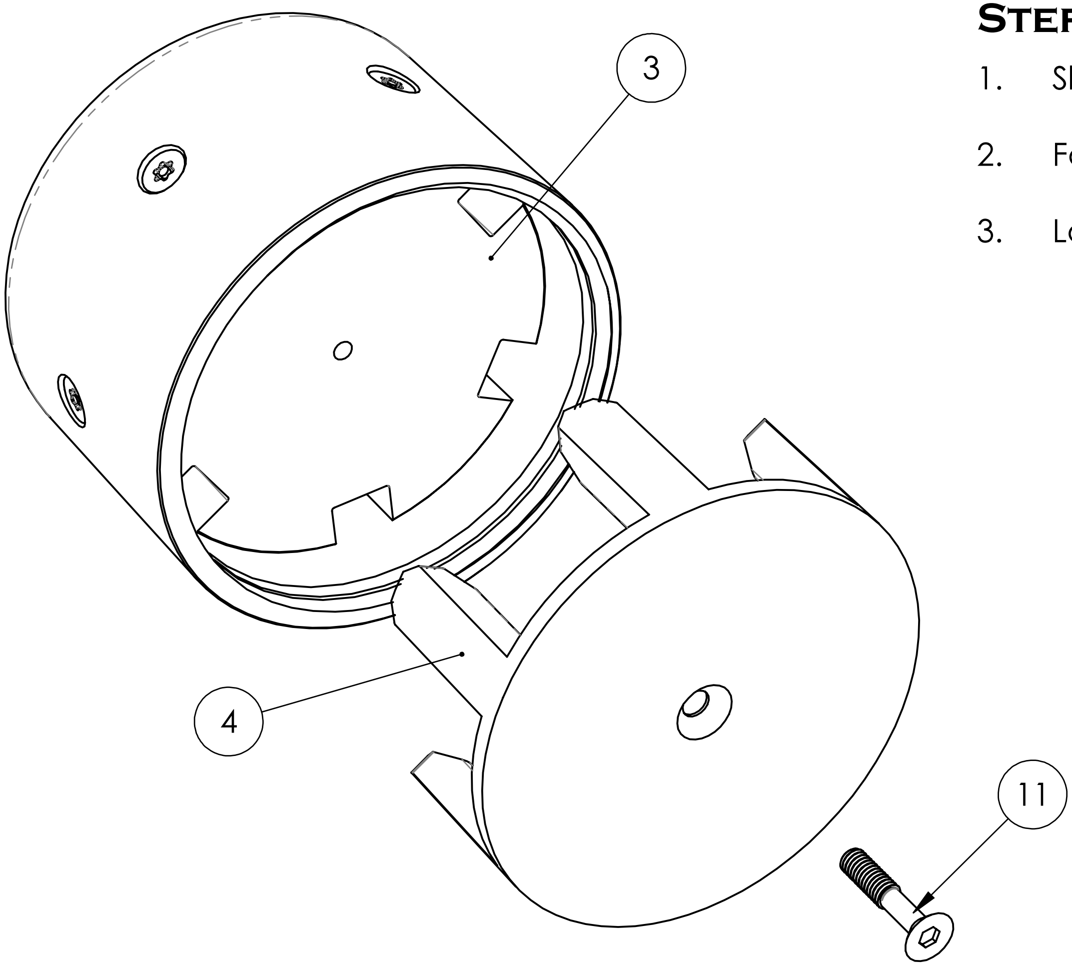
1. Rotate 2 until parallel with notch in 1
2. Slide 9 over assembly, pushing down 2
3. Fasten 9 with 6x 10
4. Remove 1 and 2





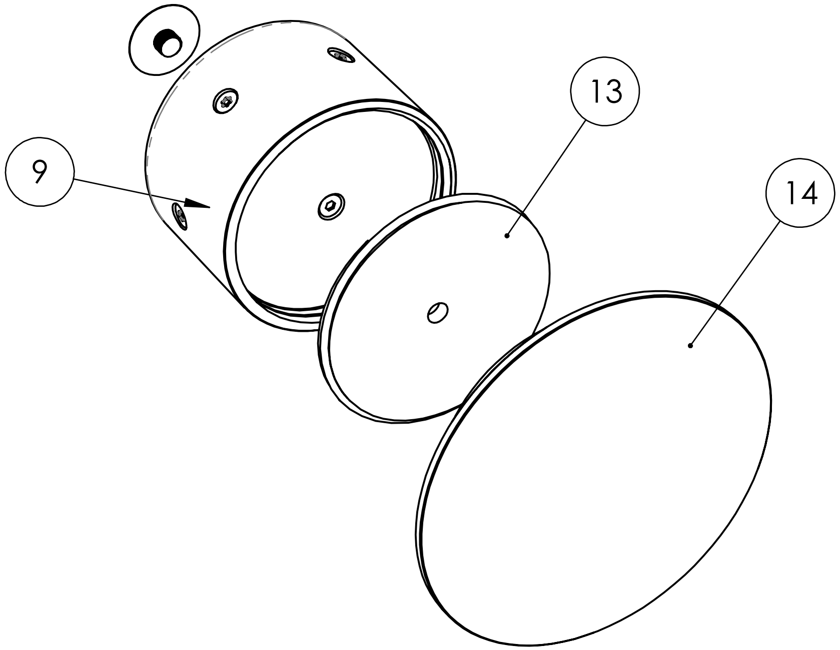






Step 4

1. Slot 4 into 3
2. Fasten with 11
3. Load 11 to **X** Nm



Step 5

1. Screw 12 into 8
2. Screw 14 into 13
3. Screw 13 into 9