MMME2044 Individual Design project

TRASL Individual Design CDR Report

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Executive summary

To initially engage the two halves of the hirth serration coupling, spring discs have been used when where 3-disc springs set are in parallel, and 5 sets are in series to provide the axial force needed to engage the couplings. Then to disengage the coupling hydraulic fluid has been used which empties into a pocket within the casing which will cause compressing forces to act on the piston, allowing the piston and therefore hirth serration coupling to move away from the third lock. The layout of the discs spring has been designed to allow a clearance distance of 4.2 mm to allow full detachment of the hirth couplings. Finally, to resist the maximum torque a female and male spline shaft have been used, calculations have been performed to show that these two components will not fail under the torsional shear stresses and axial stresses acting on them. From the calculation The max shear stress relating to the 1.5 safety factor is far higher than the shear stress experienced by the splines. Calculations were also performed to find the minimum thickness of the casing that will a safety factor of 1.5, the set thickness of the casing has been set higher than this minimum value ensuring the function of the part is achieved. The main material used in the design is annealed steel as its posses' valuable properties like high yield strength, toughness and easily machinable, all these properties will ensure the function of the part will be achieved while reducing effects like wear and tear and meeting requirements like the weight being under 3.5 Kg. The second main material is alumina because as well having a good yield strength its also lightweight so in terms of the whole system the desired weight will be achieved. Furthermore, just like the annealed steel it also helps reduce effects like wear and tear due to the surface finish of the material.

To assemble the parts turning has been used to build the piston whereas investment casting has been used to make all the other parts, both methods are suitable as they are both cheap and ensure accuracy of the final dimensions of the parts and very little machining is needed after the part has been finished. For the whole system the desired fasteners have been used to ensure the clients requirements have been met.

Design rationale and engineering

Compliance statement

No	Requirement	Outcome

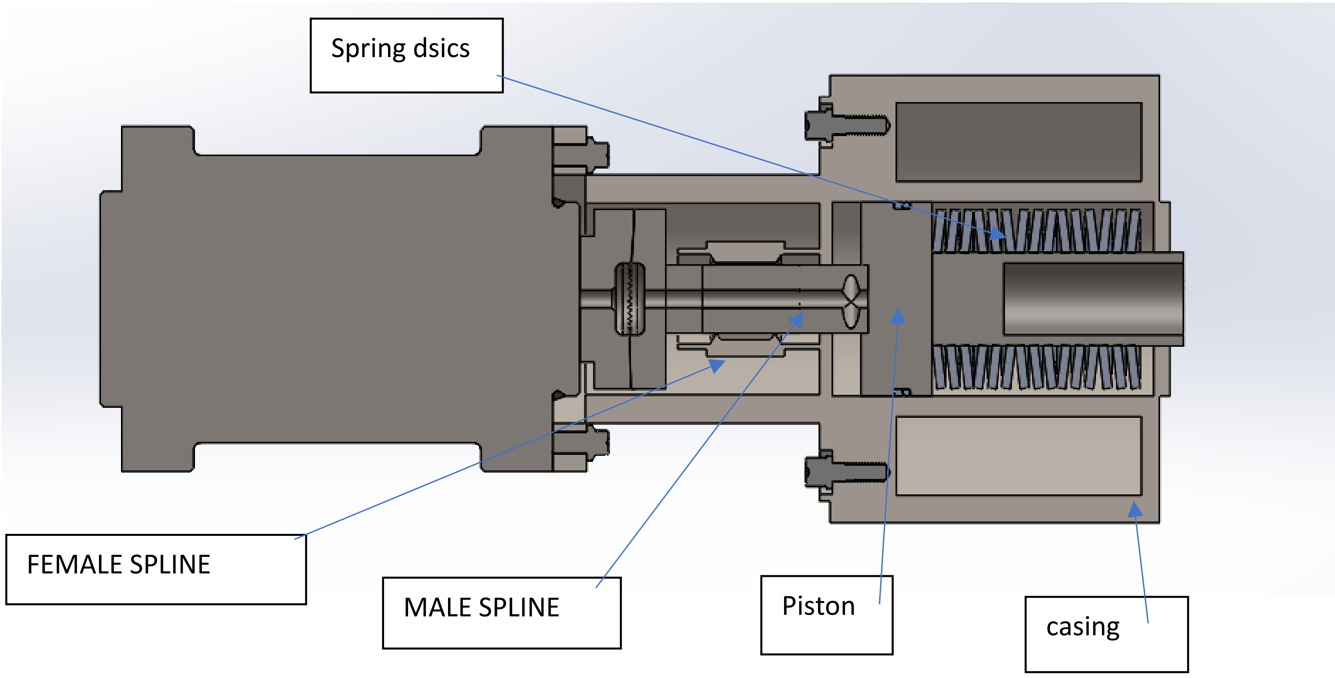
1	The TRASL must be common and fit both locking actuators and the third lock.	Dimensions of the part TRASL have been set to match the relevant dimensions of the third lock ensuring the fit is tight and common.
2	The maximum torque in the failure case is 120 Nm.	The male and female spline have been designed to resist the max torque
3	The hydraulic supply is 13.79 MPa (2000 psi) in all three hydraulic systems.	The thickness of the casing has been determined so that it withstand the hydraulic pressure
5	The total mass of the TRASL, excluding fluids, shall not exceed 3.5 kg with a target of weight of 3 kg.	The total mass of the TRASL is 3.10 Kg which is under 3.5 Kg
6	Reserve stress factors shall not be less than 1.5.	All reserves factors of the deigned parts are equal to or more than 1.5
7	For maintenance purposes the unit shall be secured to the Locking Actuators and Third Lock using M6 x 20 Aerospace Waspalloy fasteners.	The desired fasteners have been used for maintenance purposes
8	The minimum size of any other fasteners shall be M5.	The minimum fastener used is M5
9	There shall be appropriate means for sealing the TRASL with hydraulic supply of 13.79 MPs (2000psi) pressure.	An O ring and backup ring have been used to ensure complete sealing
10	Considerations are given to appropriate design features for selected manufactured processes.	All appropriate design features have been set and shown for certain manufacture processes
11	There is an established procedure for the assembly of the TRASL unit.	The assembly of the TRASL has been established in a methodical way

Morphology chart

Function	solution			
Resist the maximum applied torque	circumferential friction	radial friction	axial friction	axial friction
	band brake	drum brake	cone clutch	plate clutch
engage the lock	mechanical	mechanical	mechanical	electrical
	coil spring	disc spring	leaf spring	magnetic
disengage the lock	mechanical	mechanical	Electrical	hydraulic
	lever cam	disc spring	Magnetic	Hydraulic fluid

The solutions highlighted in red are the most suitable methods for the corresponding function

Design rationale



To engage the hirth couplings disc have been used which have been compressed to 15% of its height to provide an adequate axial force. To disengage the hirth couplings hydraulic fluid will flow through the 5 mm hole in the male spline shaft then empty into the pocket of space within the casing and compress the piston to move the piston horizontally. During this process the discs will compress to 75% of its total height. This will allow the hirth serration coupling attached to the m ale spline to move 4.2 mm in the opposite direction. The male and female spline will resist the maximum torque as the female spline will not be able to rotate

Calculations

Updated torque and force calculations

Torque and force calculations

$F_T = \frac{T}{R_m}$

$Where, \quad T = torque(Nm), F_T = Force\ due\ to\ torque\ (N), R_m = Mean\ radius\ (m)$

$F_a = KF_T \tan(\frac{\pi}{6})$

$where, \quad K = K\ factor, F_a = axial\ force\ (N)$

$$F_T = \frac{120}{0.0175} = 6857.14\ N\ \therefore\ F_a = 1.5 \times 6857.14 \times \tan\left(\frac{\pi}{6}\right) = 5938.46\ N$$

Selection of Belleville washers (disc springs) or other springs to lock and unlock the TRASL

Disc spring calculations

To engage the lock

Assuming 15% compression of a disc spring then the deflection will be

$$S1 = 0.15Ho = 0.21mm$$

The axial force generated per disc spring would therefore be 2,154 N. This is lower than the required force of 5938.46 N hence having 3 of these discs in parallel will be needed to engage the lock.

$$F_{a1} = 2154 \times 3 = 6462\ N \therefore\ F_{a1} > F_a$$

To disengage the lock

Assuming 75% compression of a disc then the deflection will be

$$S2 = 0.75Ho = 1.05mm$$

The axial force generated per disc spring would therefore be 9063 N. Having 3 of these discs in parallel will result in an axial force of

$$F_{a2} = 9063 \times 3 = 27189\ N$$

The change in deflection to fully disengage the lock is 4.2 mm and the change in deflection of the 3 discs in parallel is, $\Delta S = 22 - S1 = 0.6Ho = 0.84\ mm$

Therefore, the number springs needed in series is

$$N = \frac{4.2}{0.84} = 5$$

Sizing of the piston and cylinder

Sizing of piston head

$$A = \frac{F_{a1}}{P} = \frac{6462}{13.8 \times 10^6} = 4.683 \times 10^{-4}\ m^2$$

Hence the minimum area of the piston head is $4.683 \times 10^{-4}\ m^2$, the corresponding diameter is

$$D = \sqrt{\frac{4A}{\pi}} = 0.0244\ m = 24.4\ mm$$

Therefore, the piston head diameter must be ≥ 24.4 mm. The piston head diameter I have chosen is 54 mm.

Sizing of piston rod

$$A = \frac{F_{a2}}{P} = \frac{27189}{13.8 \times 10^6} = 1.97 \times 10^{-3} m^2$$

$$A = A_{piston} - (A_{shaft} + A_{oil\ hole})$$

$$A_{oil\ hole} = \pi \times (2.5 \times 10^{-3})^2 = 1.96 \times 10^{-5} m^2$$

The diameter of the piston is equal to the diameter of the casing which is equal to 54mm therefore

$$A_{piston} = \pi \times (27 \times 10^{-3})^2 = 2.29 \times 10^{-3} m^2$$

$$A_{shaft} = A_{pisotn} - A + A_{oil\ hole} = 3.396 \times 10^{-4} m^2$$

$$D_{shaft} = \sqrt{\frac{4 \times A_{shaft}}{\pi}} = 21mm$$

Thus, the diameter of the shaft, must be equal to or less than 21mm. From this I set the diameter of the shaft as 19mm.

O ring calculations [1]

The dimensions of the O ring can be found using (reference) table and knowing the diameter of the piston.

Since the diameter of the piston is 54 mm, the following quantities apply to the O ring:

- A, cross sectional diameter = 2.4 mm^2
- F, radial depth, Max = 2.09mm and min= 1.97mm
- ${E_o}^{+0.2}$, Groove width= 3.2mm
- G(max), Total dimensional clearance= 0.14mm
- C,chamfer= 0.6mm
- R, Max radius= 0.5mm
- Internal diameter= 49.6$\pm 0.3\ mm$
- Section diameter= 2.4$\pm 0.08 mm$

Back up ring [2]

For the backup ring, the dimensions used were adjusted from the o ring dimensions stated above.

Adjustments:

- Groove width = 4.8 mm
- Chamfer=2.2mm
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Reserve factor of designed parts

Reserve factor for Piston head

Assuming the piston head is made of annealed steel, the yield strength of annealed steel stated on solid works is 470 MPa

Using the tresca yield criterion

$$\tau_y = \frac{1}{2}\sigma_y \therefore\ \tau_y = \frac{470}{2} = 235\ MPa$$

With a reserve factor of 1.5 the max shear stress and max tensile stress is

$$\tau_{max} = \frac{\tau_Y}{1.5} = 156.67\ MPa$$

$$\sigma_{max} = \frac{\sigma_y}{1.5} = 313.33\ MPa$$

$$tensile\ stress, \sigma_x = \frac{F}{A} = \frac{PA}{A} = 13.79\ MPa$$

$$Max\ torsional\ shear\ stress, \tau_{yxmax} = \frac{Tr}{J}$$

$$J = \frac{\pi d^4}{32} = \frac{\pi \times (54 \times 10^{-3})^4}{32} = 8.35 \times 10^{-7} m^4 \therefore$$

$$\tau_{yx} = \frac{Tr}{J} = \frac{120 \times (27 \times 10^{-3})}{8.35 \times 10^{-7}} = 3.88\ MPa$$

$$Max\ stress, \sigma_1 = C + R$$

$$C = \frac{\sigma_x}{2} = 6.895\ MPa$$

$$R = \sqrt{(\frac{\sigma_x}{2})^2 + \tau_{yxmax}{}^2} = \sqrt{6.985^2 + 3.88^2} = 7.99\ MPa$$

$$\sigma_1 = 14.89\ MPa$$

Since σ_1 is less than σ_{max} and τ is less than τ_{max} this shows the piston head will be able to withstand the shear and tensile stress exerted onto it and will not fail.

Reserve factor for male spline shaft

Assuming the male spline shaft is made of annealed steel, the yield strength of annealed steel stated on solid works is 470 MPa

Using the tresca yield criterion and assuming a reserve factor of 1.5 the max shear stress is

$$\tau_y = \frac{1}{2}\sigma_y \therefore\ \tau_y = \frac{470}{2} = 235\ MPa$$

$$\tau_{max} = \frac{\tau_Y}{1.5} = 156.67\ MPa$$

The shear force experienced by the teeth of the male spline is

$$F = \frac{T}{pitchh\ radius, r} = \frac{120}{12 \times 10^{-3}} = 10\ KN$$

Thus, the shear force acting on the teeth of the male spline due to this force is

$$\tau = \frac{F}{\pi \times (9.5 \times 10^{-3})^2} = 35.27\ MPa$$

Since τ is less than τ_{max} the male spline will be able to withstand the shear force exerted on it

Reserve factor for shear pin

Assuming the shear pin is made of AISI 34130 steel as stated on solid works, then the shear force experienced by the shear pin is

$$F = \frac{T}{r} = \frac{120}{0.04} = 3\ KN$$

Therefore, the shear force acting on the pin due to this force is

$$\tau = \frac{F}{\pi \times (2.995 \times 10^{-3})^2} = 106.46\ MPa$$

Using the tresca yield criterion

$$\tau_y = \frac{1}{2}\sigma_y \therefore\ \tau_y = \frac{460}{2} = 230\ MPa$$

$$\tau_{max} = \frac{\tau_Y}{1.5} = 153\ MPa$$

Therefore since τ is less than τ_{max} the shear pin can withstand the shear stress exerted on it and will not fail.

Reserve factor of casing

Assuming the casing is made of AISI 4340 annealed steel, the yield strength of this material stated on solid works is 470 MPa

Using the tresca yield criterion

$$\tau = \frac{1}{2}\sigma_y \therefore \ \tau = \frac{470}{2} = 235\ MPa$$

With a reserve factor of 1.5 the maximum tensile stress experienced by the casing is

$$\sigma_{max} = \frac{\sigma_y}{1.5} = 313.33\ MPa$$

Whereas the maximum shear stress experienced by the casing is

$$\tau_{max} = \frac{\tau}{1.5} = 156.67\ MPa$$

$$Hoop\ axial\ stress, \sigma_\theta = \frac{Pr}{t} \therefore t = \frac{Pr}{\sigma_\theta} = \frac{13.79 \times 10^6 \times 62.24 \times 10^{-3}}{\sigma_{max}} = 2.74\ mm$$

Therefore, the minimum thickness of the casing must be equal to or more than 2.74 mm in order to have a reserve factor of 1.5 or more.

The torsional shear stress experience by the casing is

$$J = \frac{\pi((62.24 \times 10^{-3})^4 - (27 \times 10^{-3})^4)}{32} = 1.42\ \mu m^4$$

$$\tau_{yx} = \frac{Tr}{J} = \frac{120 \times 62.24 \times 10^{-3}}{J} = 5.26\ MPa$$

Since τ_{yx} is less than τ_{max} this shows the casing will be able to withstand the shear stress exerted on the casing.

Tightening torque of fasteners

M6 Fastener

$$\mathrm{P} = 27000, \mathrm{N} = 6, \mathrm{K} = 0.2$$

$$F_i = 0.75 \times F_p = 0.75 \times 11600 = 8700$$

$$P_0 = 6 \times 8700 \times \frac{(70.2 \times 10^3) + (211.3 \times 10^3)}{211.3 \times 10^3} = 69542.4$$

$$N_0 = \frac{P_O}{P} = 2.57$$

Therefore, the tightening torque is

$T = 0.2 \times 8700 \times 6 = 10440\ Nm$

M5 fastener

$$\mathrm{P} = 27000, \mathrm{N} = 6$$

$$F_i = 0.75 \times F_p = 0.75 \times 8230 = 6172.5$$

$$P_0 = 6 \times 6172.5 \times \frac{(70.2 \times 10^3) + (211.3 \times 10^3)}{211.3 \times 10^3} = 49339.10$$

$$N_0 = \frac{P_O}{P} = 1.827$$

Therefore, the tightening torque is
$T = 0.2 \times 617.5 \times 5 = 6172.5\ Nm$

Material selection

Piston head and shaft
The material of the piston head and shaft will be annealed steel, this is due to the advantageous properties of this material for example toughness, yield strength and machinability of this metal [3]. This would be useful in making this piston as the piston would be under constant compression therefore would need to be made of a material that could withstand this stress in order to avoid failure of the system. Furthermore, the machining of the piston would be considered easier compared to other parts thus the fact that annealed steel is easy to machine would mean the production of the part would be smooth. Finally, the wear resistance of annealed steel is very good making it again optimal in the material of the piston as the piston would have to function over several cycles.

Process of manufacturing
The piston head and shaft will be manufactured via turning which seems the most practical method as not only is it cheap to produce the part this way, but it also produces very accurate dimensions when used correctly due to the cylindrical shape of the part.

Casing and female spline
The material of the female spline of and casing will be aluminium, this is due to the durability of aluminium being useful in the function of the female spline as the female spline will be in constant contact with its counterpart therefore allowing forces like friction to arise which overtime will wear the part down, fortunately by using aluminium this effect will be minimised thus increasing the life span of the female spline. Furthermore, aluminium is considered lightweight thus proving useful for the casing as the total weight of the TRASL is mainly made of the weight of the casing thus using aluminium will minimise the wight of the casing allowing the target weight specified in the requirements to be achieved. Finally, the casting and machinability of the aluminium is also very good therefore the production of these two parts would be less difficult.

Process of manufacturing
The casing and female spline will be manufactured via investment casting [4], this method would be most suitable as both parts are more complex in terms of shape when compared to for example the piston, thus investment casting would allow an accurate production of the part with a great surface finish and minimal machining after the parts have been casted.

Coupling
The material of the hirth serration coupling will be annealed steel because as previously said the yield strength and toughness of annealed steel is very good. These two properties would be advantageous in the material of the hirth serration coupling due to its function. The coupling is always in constant contact

with its other half either by an axial stress or when torque is applied to it therefore forces like friction will arise and wear the part down, however by using annealed steel this effect will be minimised thus prolonging the life span of the hirth serration coupling.

Process of manufacturing

The coupling will be manufactured again via investment casting, this method would be most suitable as this part is again more complex in terms of shape thus investment casting would allow an accurate and top-quality part to be produced.

Male spline

The material of the male spline will also be annealed steel as again the toughness and yield strength combined with the density of the material prove useful in the function of the male spline. The male spline is always in contact with the piston and sometimes with female spline thus again forces like friction will arise which contribute to the wearing down of a part. Using annealed steel will as you know reduce this effect thus allowing to male spline to function for a longer period without failure occurring.

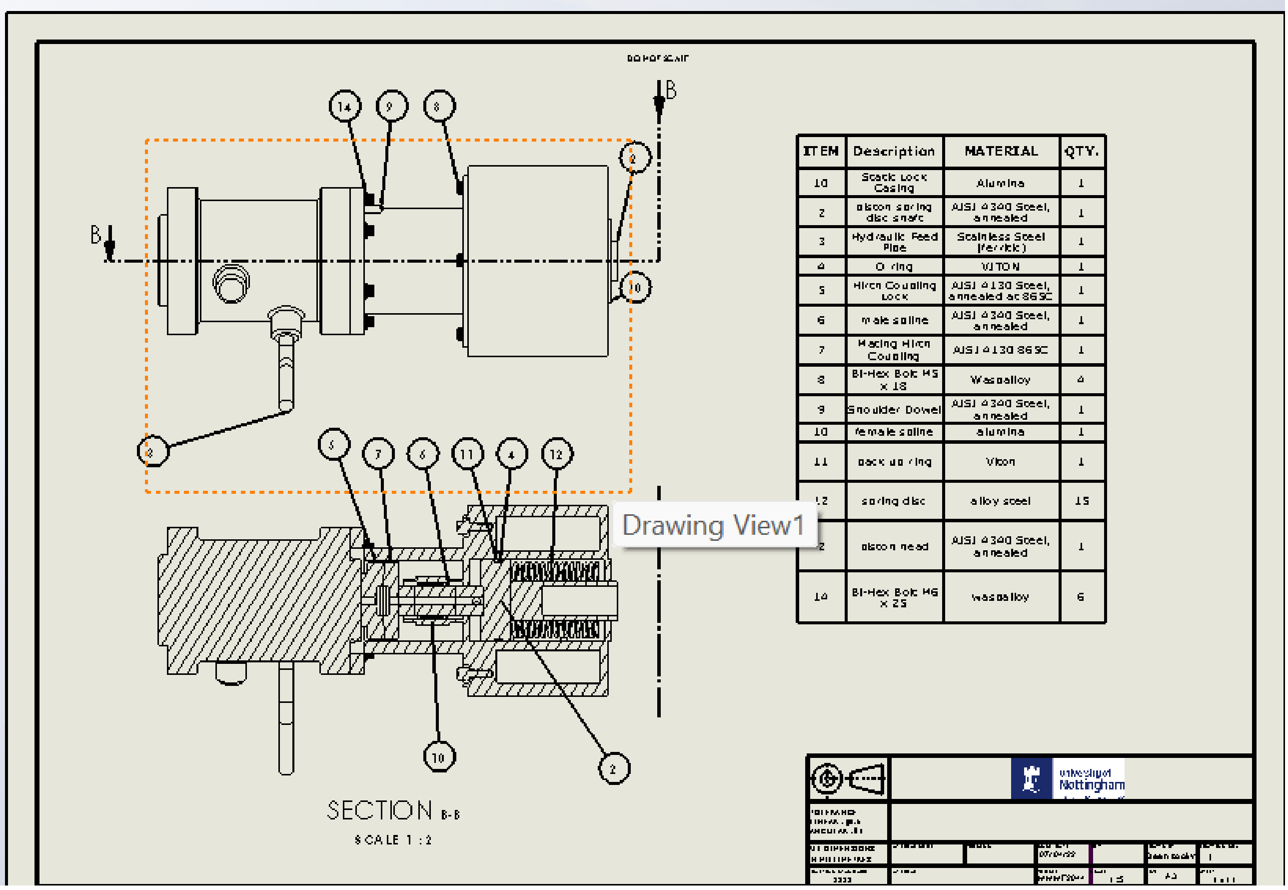
Process of manufacturing

The male spline will be manufactured via investment casting as well, this method would be most suitable because as you already know this method provides accurate and quality parts with little machining being needed after casting. These factors combined will low production volume will mean this method is also cheap to do, thus decreasing the overall cost.

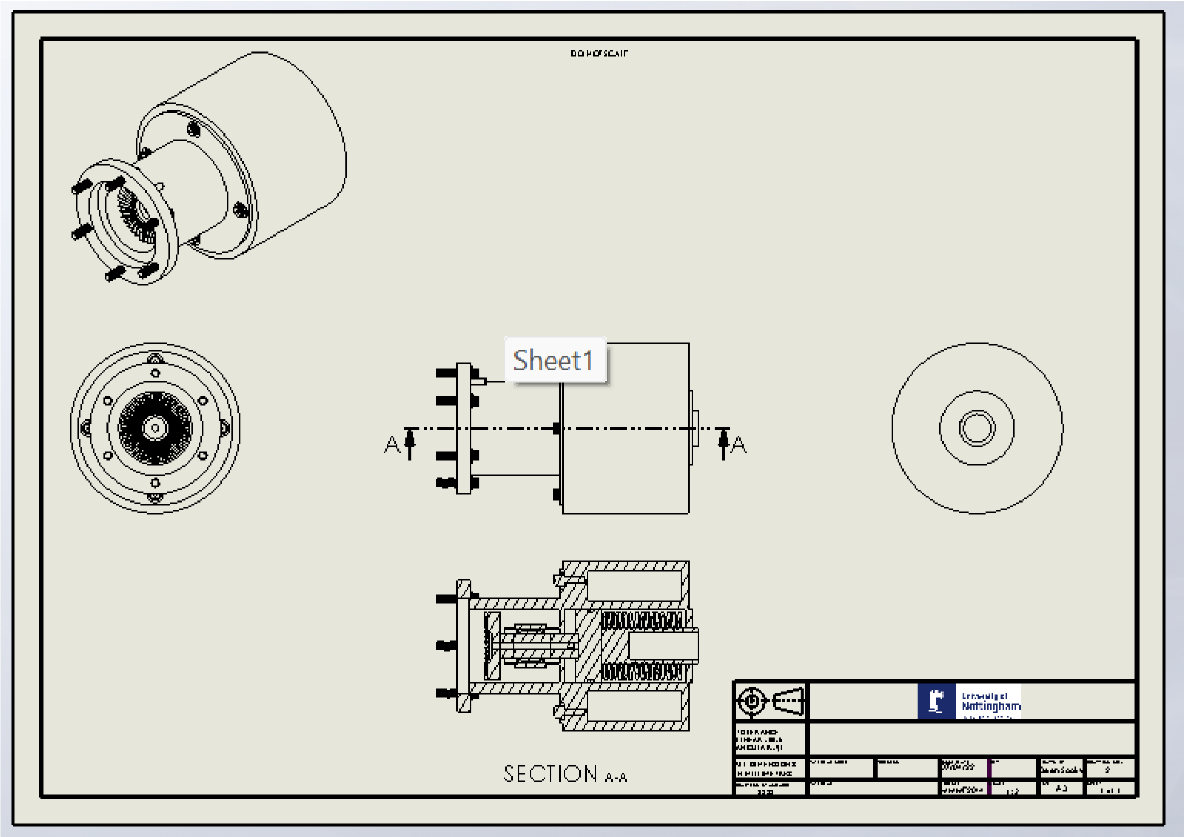
The total weight of the TRASL is 3.10 Kg which meets the requirements of being under 3.5 Kg##

GA drawings

Ga drawing of system engaged

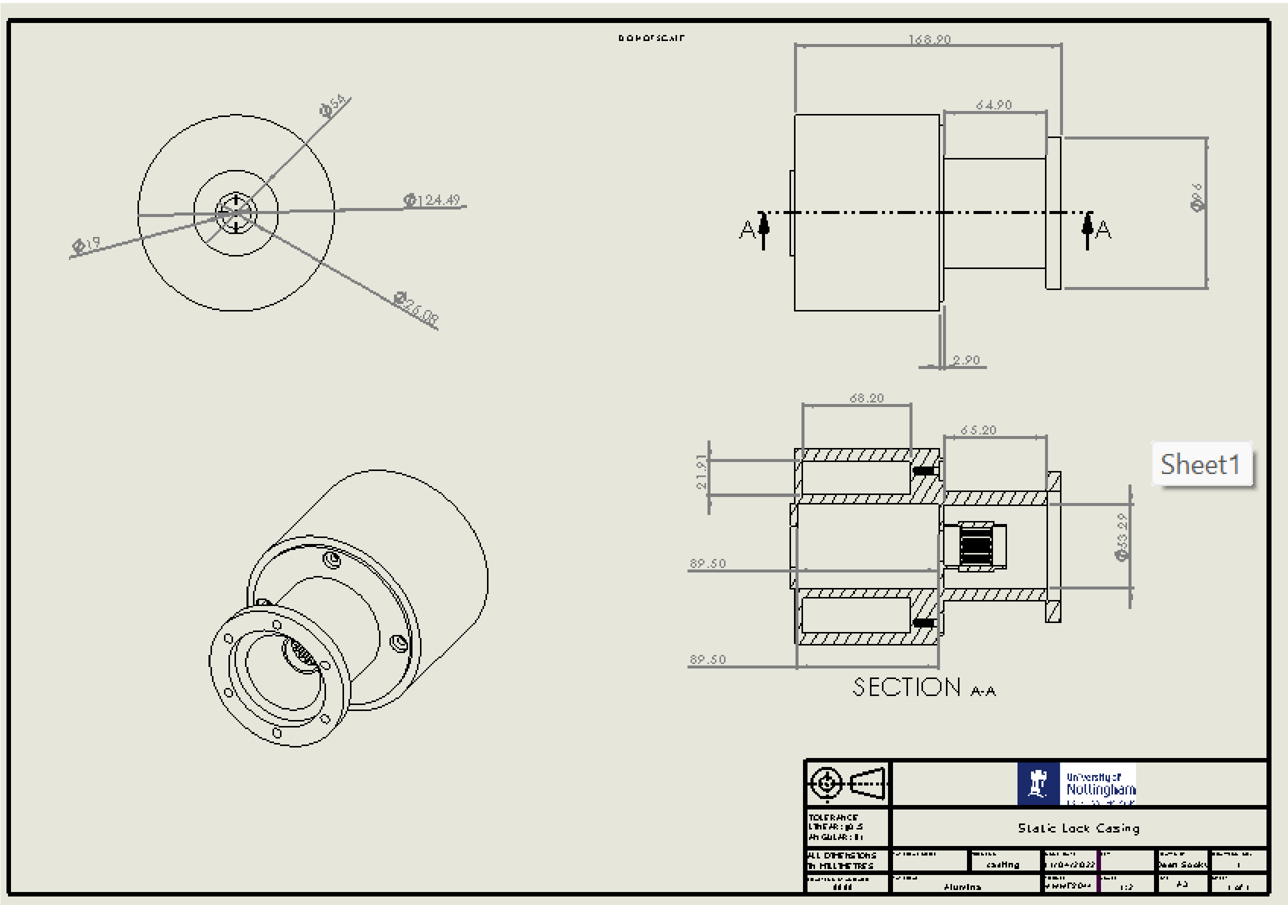


GA drawing TRASL

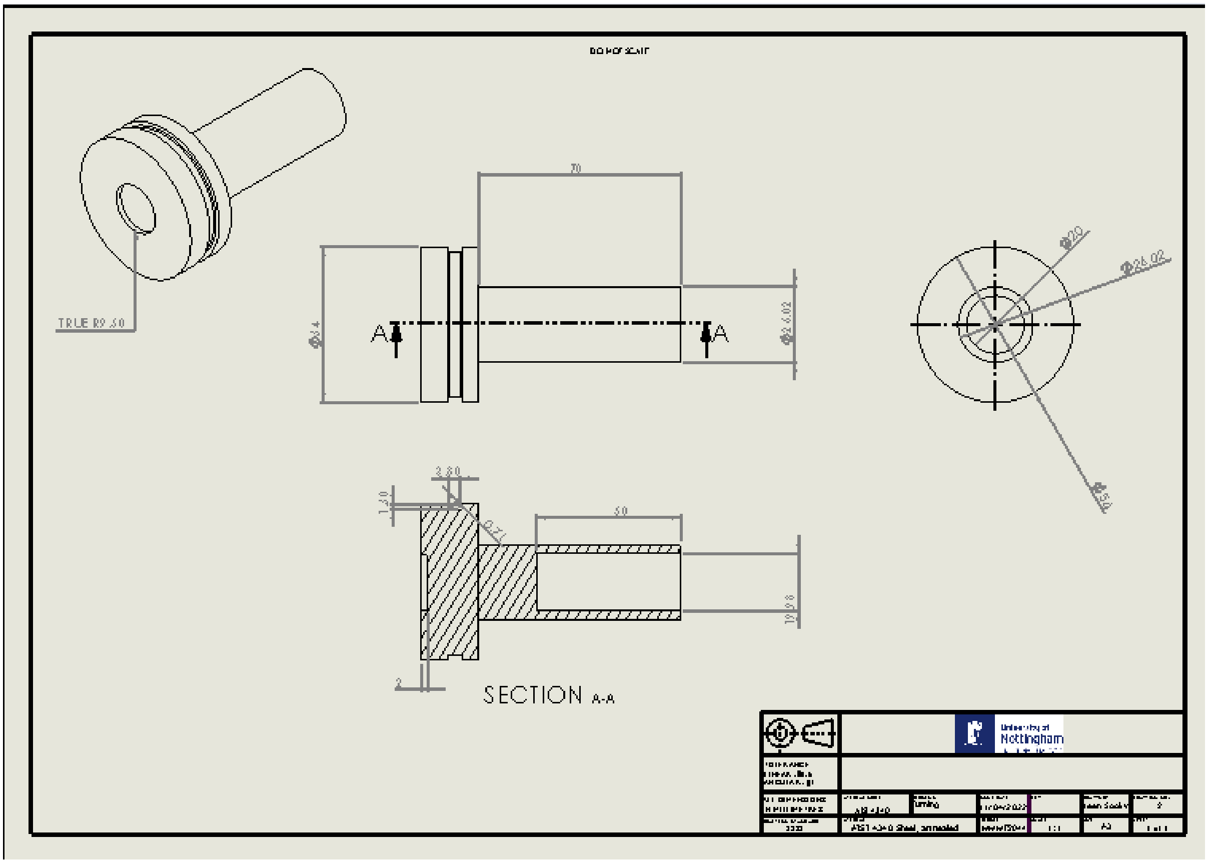


Detail drawings

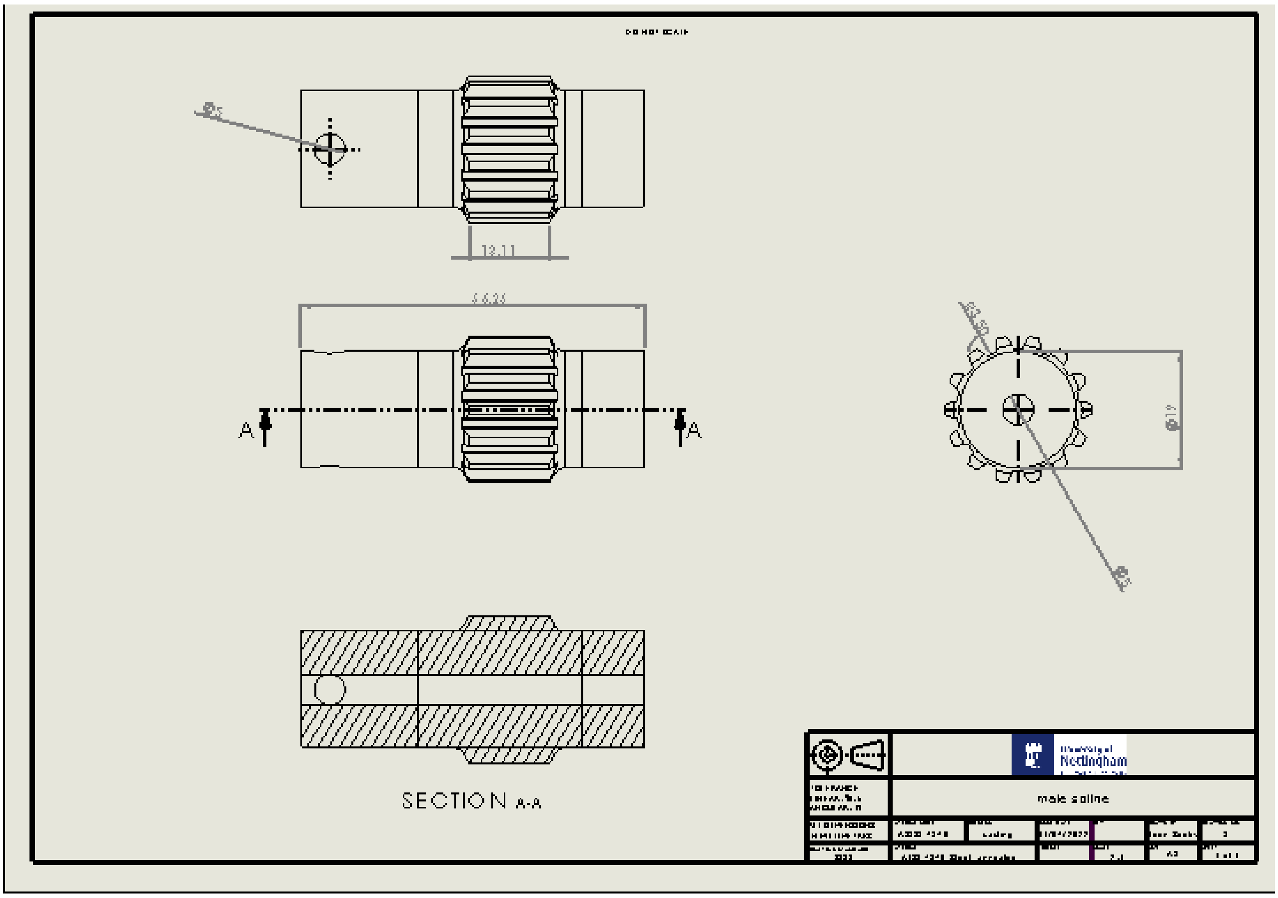
Detail drawing of female spline and casing



Detail drawing of piston



Detail drawing of male spline shaft



References

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