MECH2003, Mechanical Engineering Design

Design of Simple Rotating Systems and Tolerances

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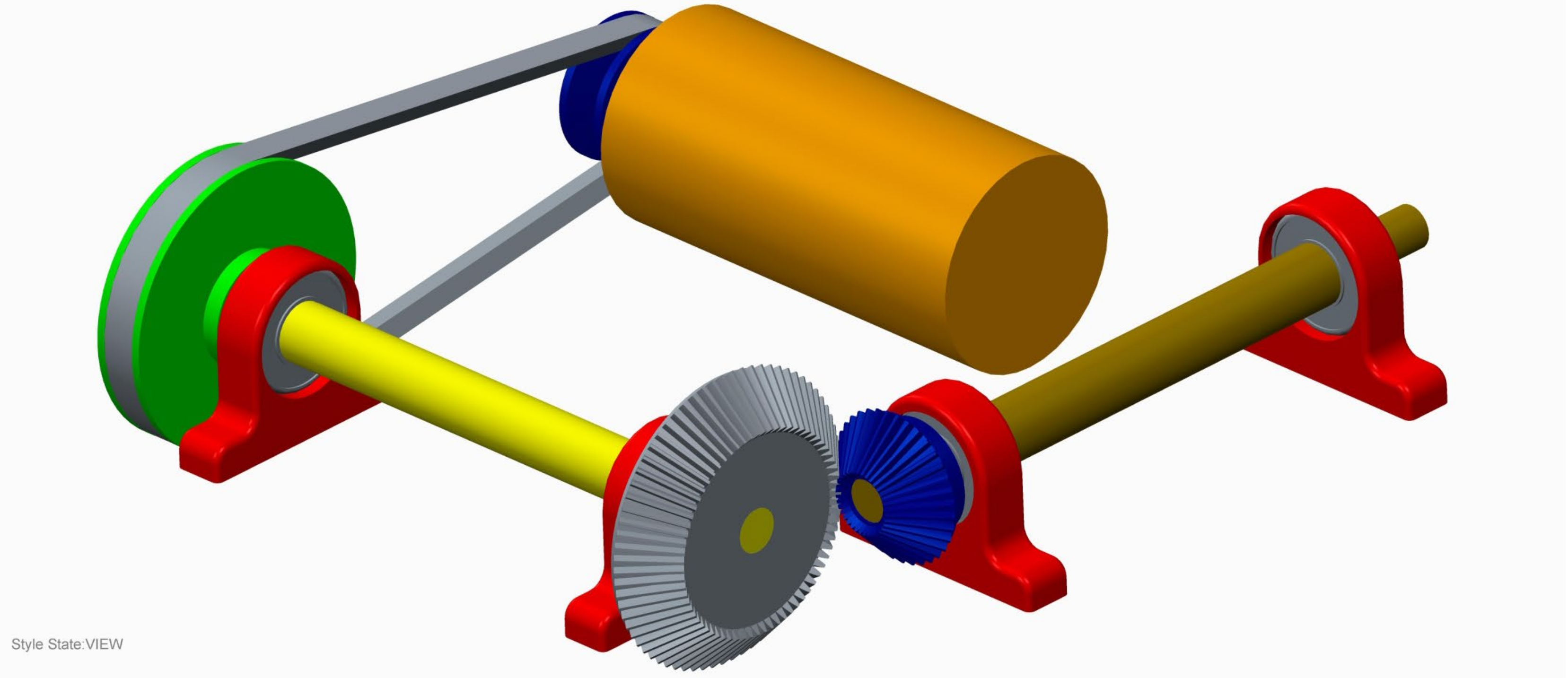
Design Problem – Rotating Shaft System

A common mechanical engineering problem is to transfer mechanical power from the location that it is created to where it is required.



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Design Problem – Rotating Shaft System

Example of a rotating shaft system include *– providing power for machines in factories…*

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Design Problem – Rotating Shaft System

Example of a rotating shaft system include *– shearing sheep…*

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Design Problem – Rotating Shaft System

Example of a rotating shaft system include *– rear wheel vehicle drive trains…*

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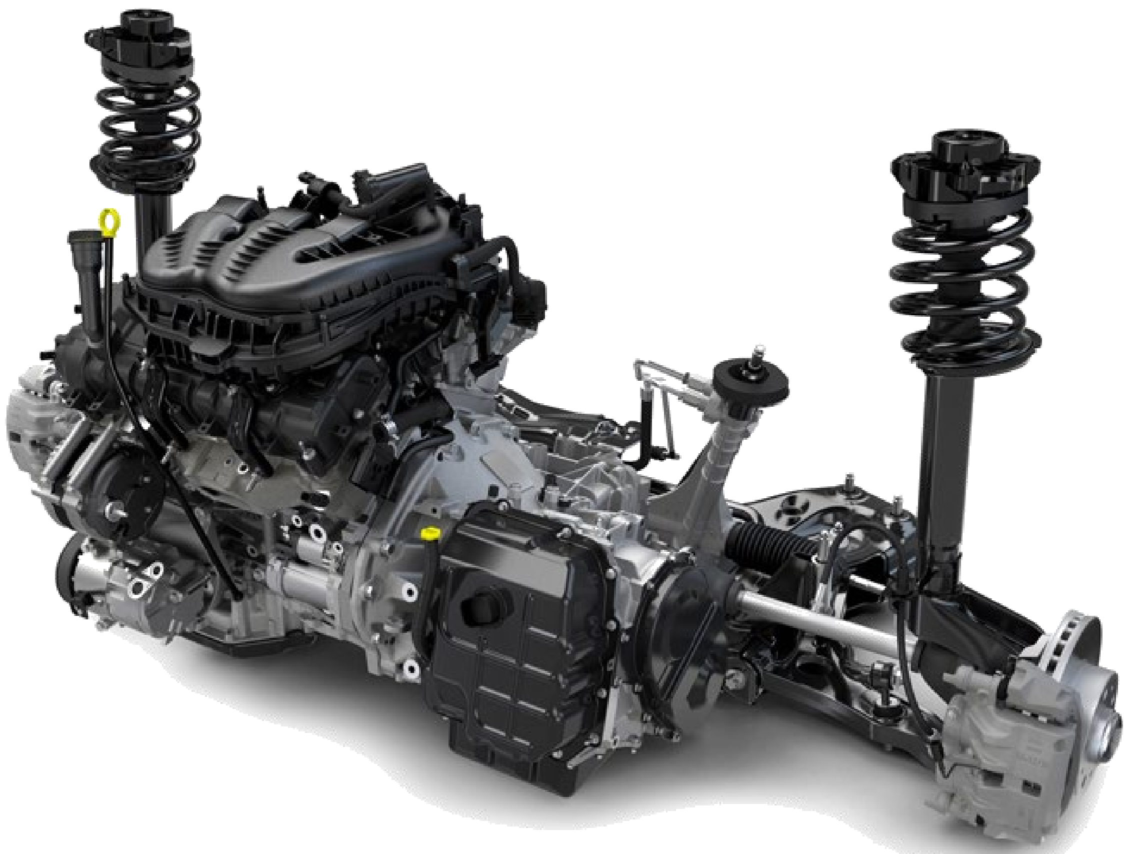
Design Problem – Rotating Shaft System

Example of a rotating shaft system include *– front wheel vehicle drive trains…*



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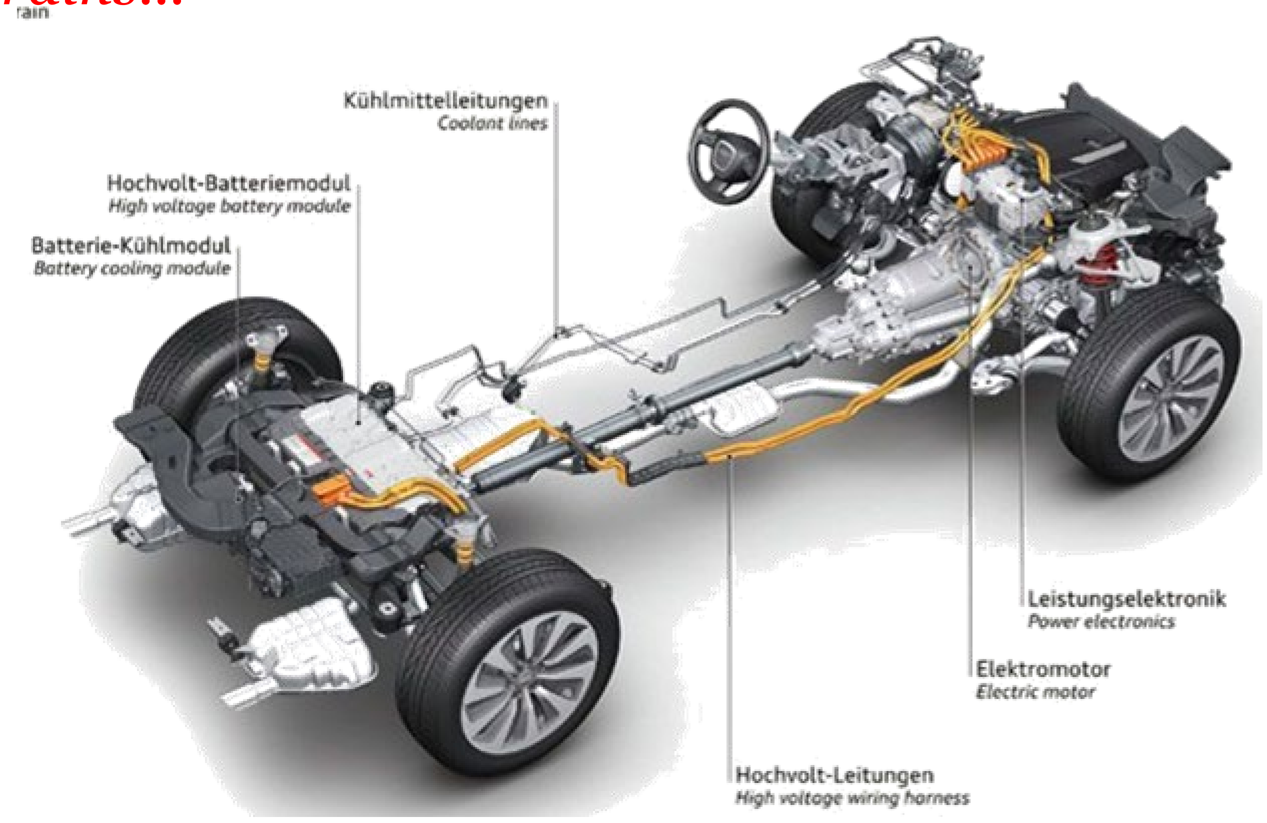
Design Problem – Rotating Shaft System

Example of a rotating shaft system include *– four wheel vehicle drive trains…*



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Design Problem – Rotating Shaft System

Example of a rotating shaft system include *– fishing reels…*

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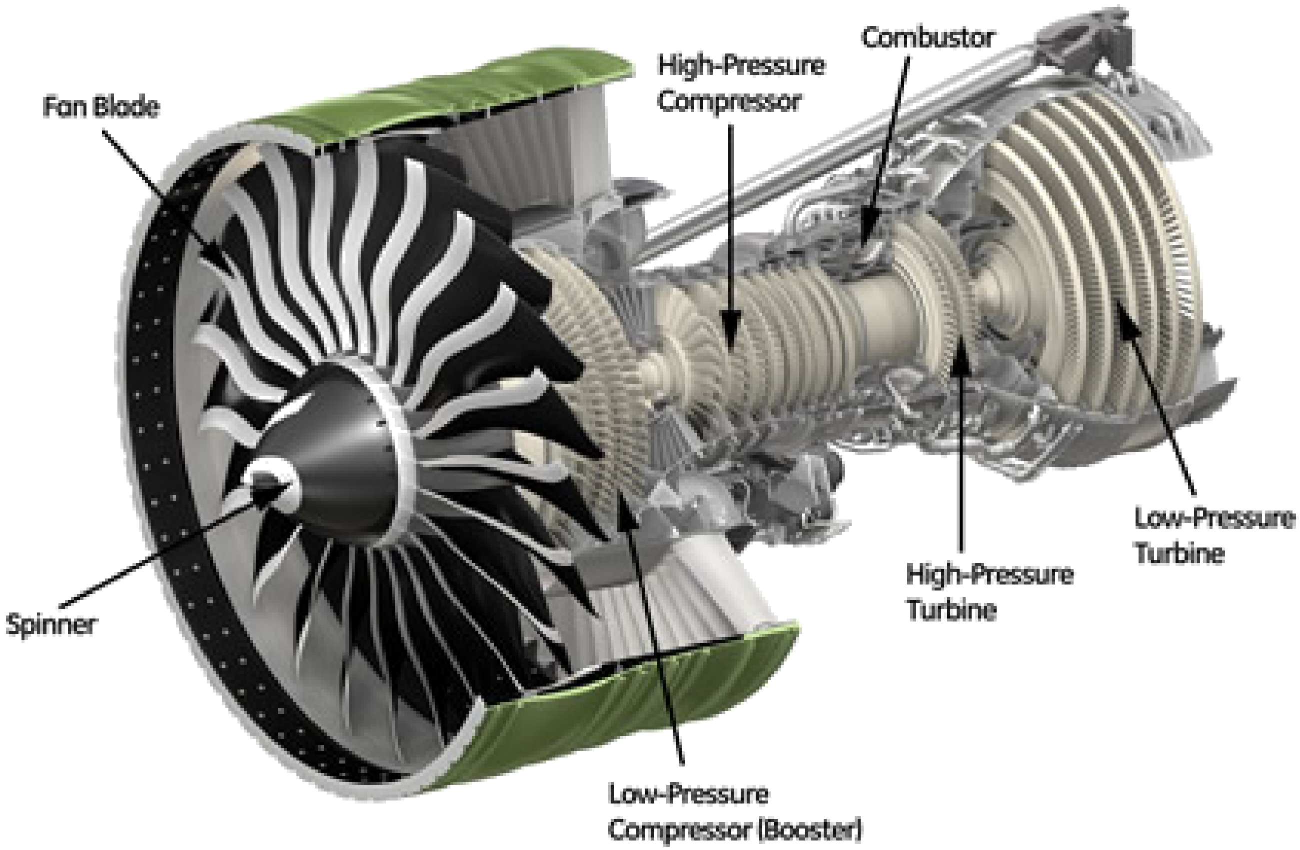
Design Problem – Rotating Shaft System

Example of a rotating shaft system include – Turbofan jet engines…



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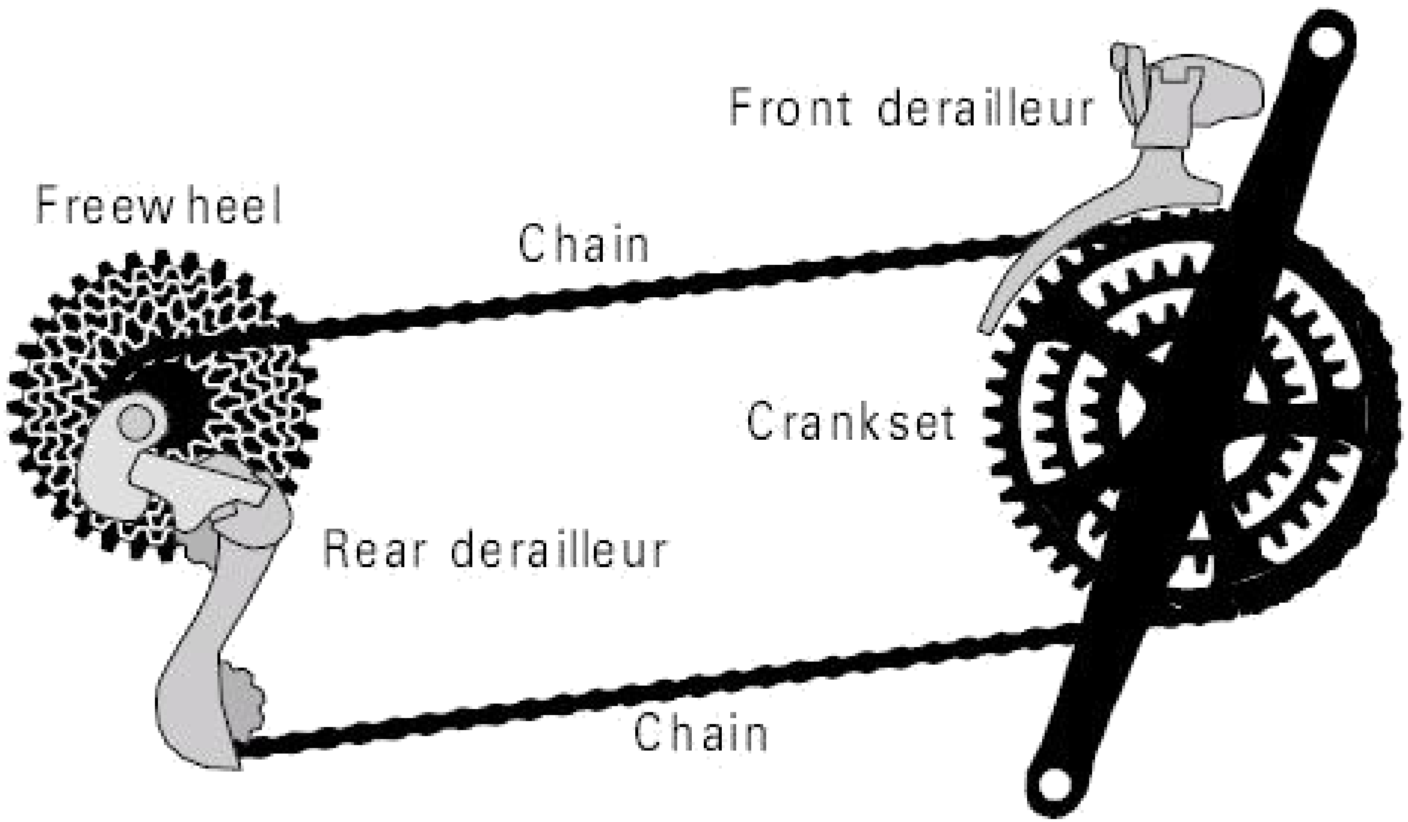
Design Problem – Rotating Shaft System

Example of a rotating shaft system include – *bicycle…*



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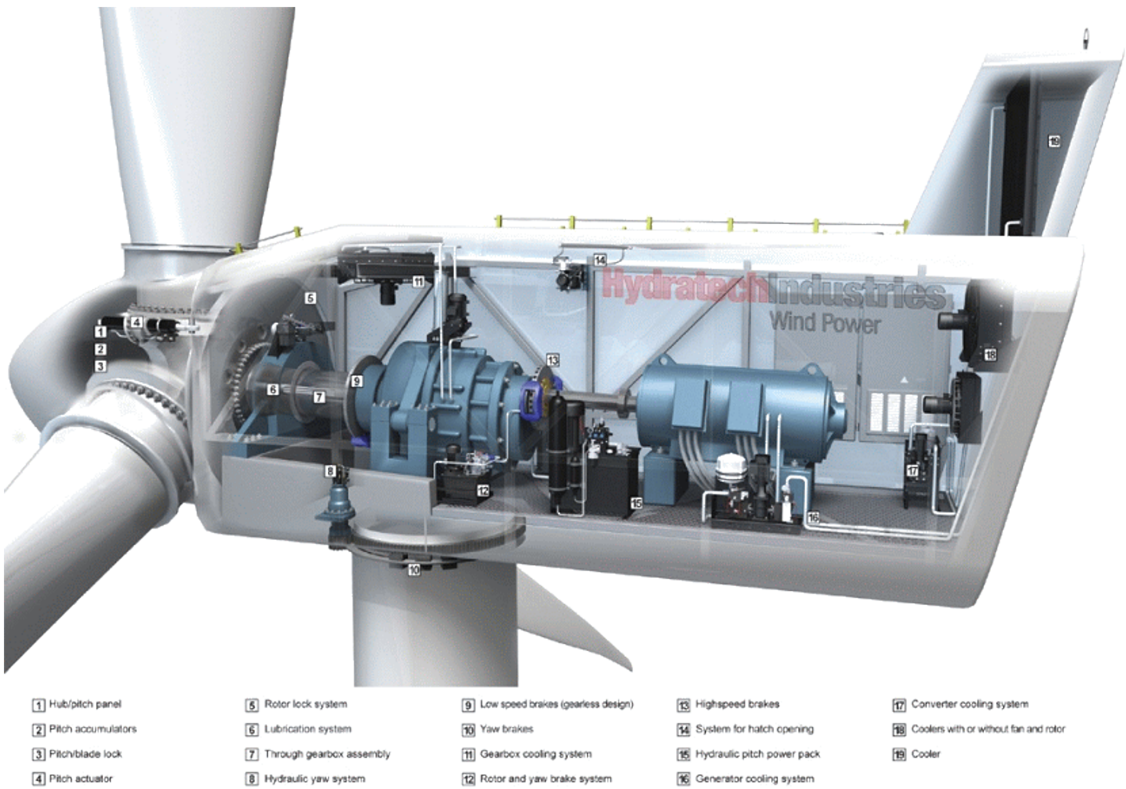
Design Problem – Rotating Shaft System

Example of a rotating shaft system include – *Wind turbine generators…*



All the mechanisms shown thus far have one thing in common, they have shafts that are rotating at a variety of angular velocities with various torques.

But somehow, rotating shafts need to be held in position without restricting their rotation. This is what bearings and bushings are required for.

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Allowing shafts to rotate

Bushings attempt to minimise the friction through careful material selection. Ball bearings minimise the friction by incorporating elements that roll instead of slide. As a result, ball bearings are far more efficient!

Bushings

Ball Bearings

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Allowing shafts to rotate - bushings

Bushings are used when cost must be minimised or the shaft rotation speed and loads perpendicular to the shaft are small. They are made of materials that are softer than the shaft so that they wear sacrificially.

Brass Bushings

Bronze Bushings

Nylon Bushings

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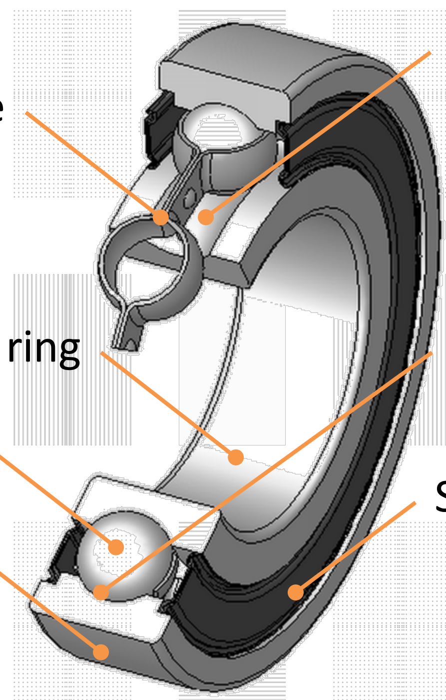
Allowing shafts to rotate - bearings

For high speed applications where the loads being carried by a shaft are also high, bearings are more suitable. Many different type exist, but they all contain similar components.



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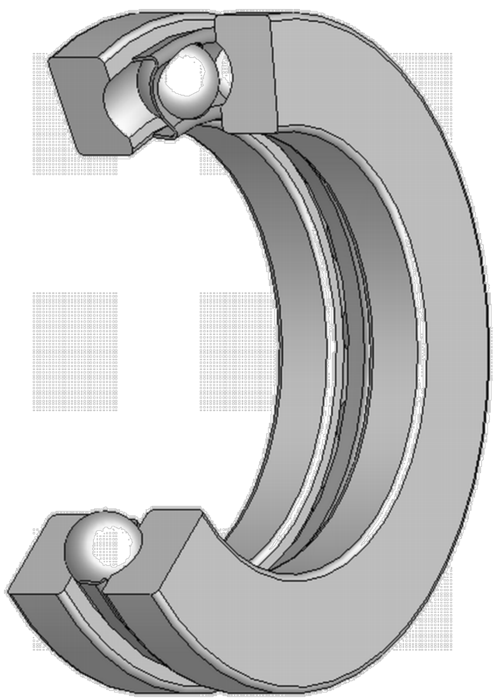
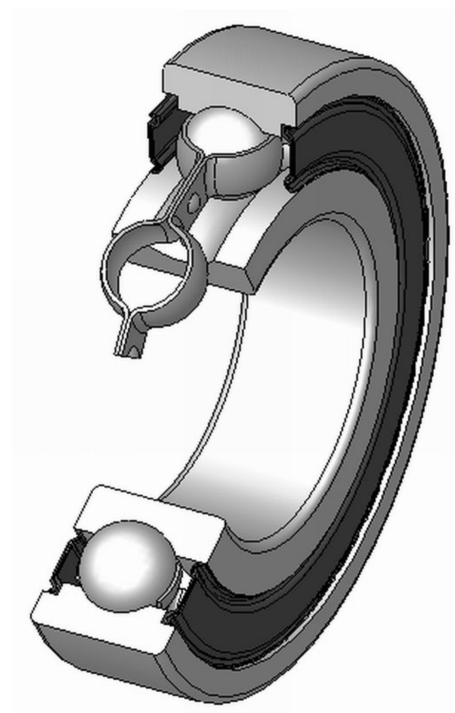
Allowing shafts to rotate - bearings

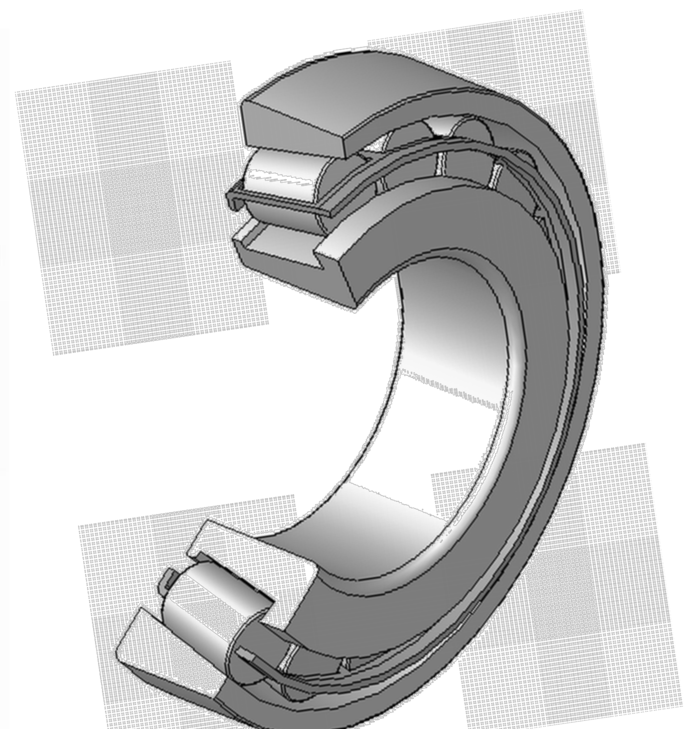
Many different bearings exist with the type required being dependent on the direction of the load being transferred.



Thrust ball bearing



Deep groove ball bearing



Tapered roller bearing

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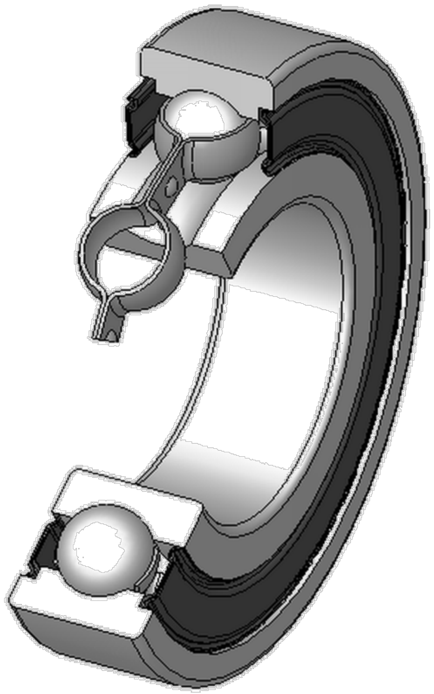
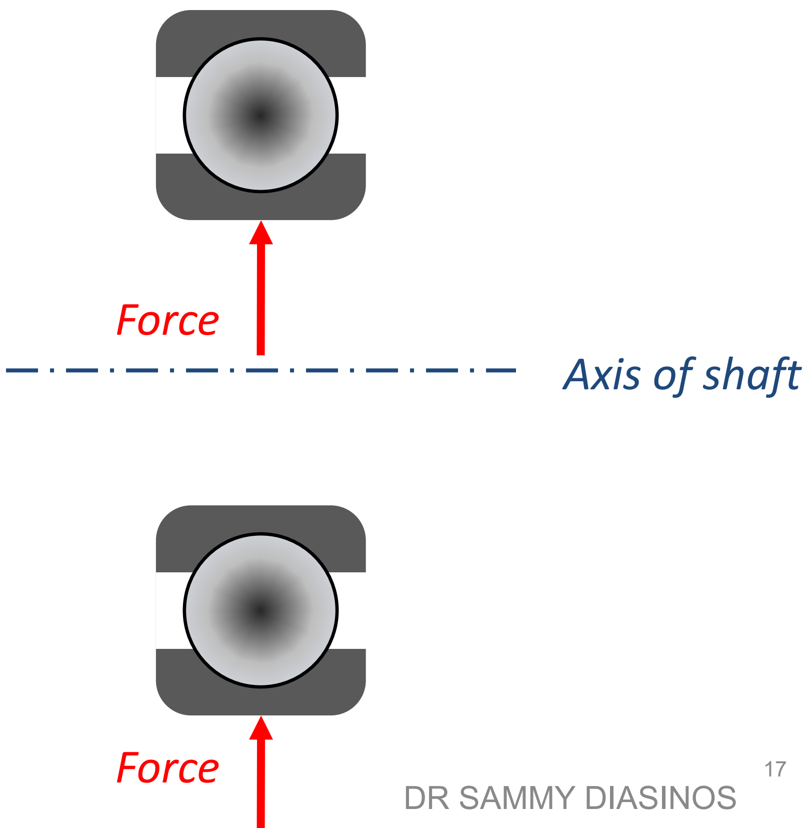
Allowing shafts to rotate - bearings

Deep groove ball bearings are designed to accommodate loads that are normal to the axis of rotation. Therefore the steel balls/rollers are supported in this direction



Deep groove ball bearing



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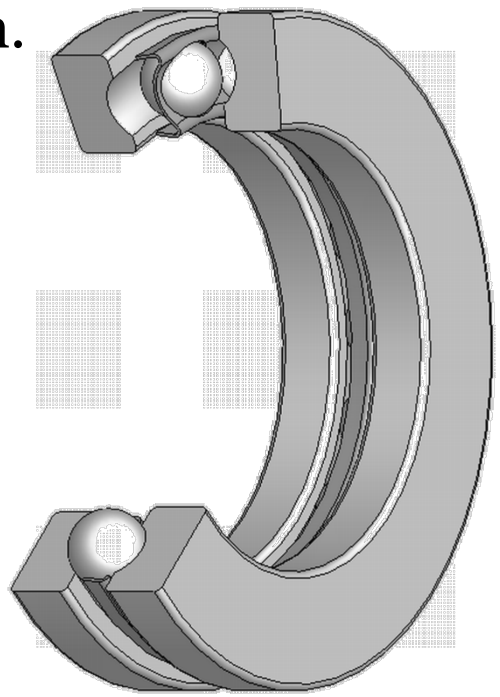
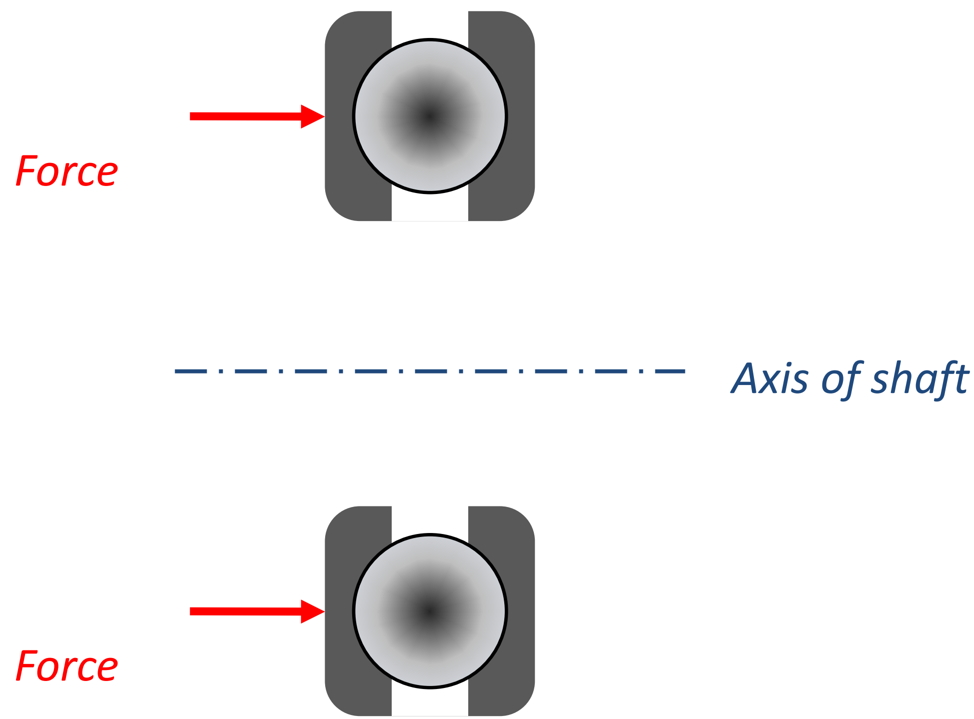
Allowing shafts to rotate - bearings

For high loads in the direction normal to the shaft, thrust bearings are most suitable. The raceways and rings are reorientated to provide support to the steel balls/rollers in this direction.



Thrust ball bearing



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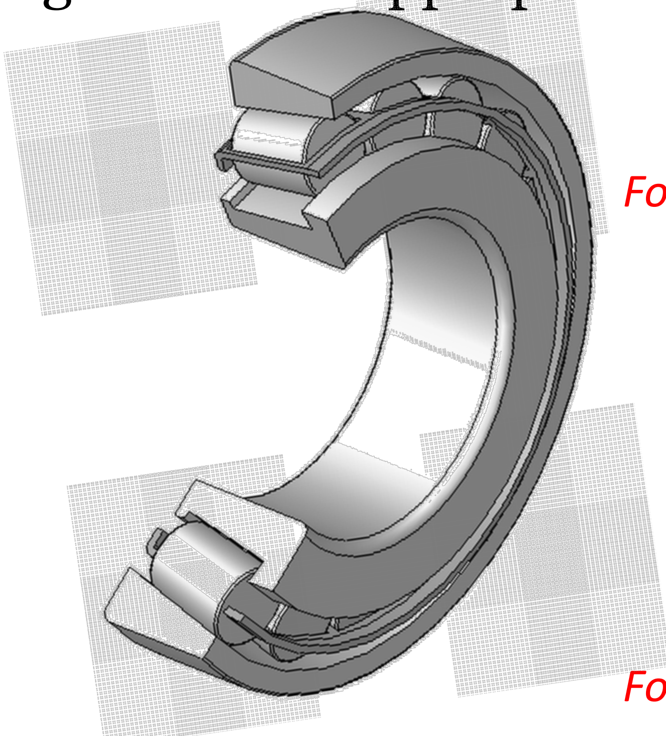
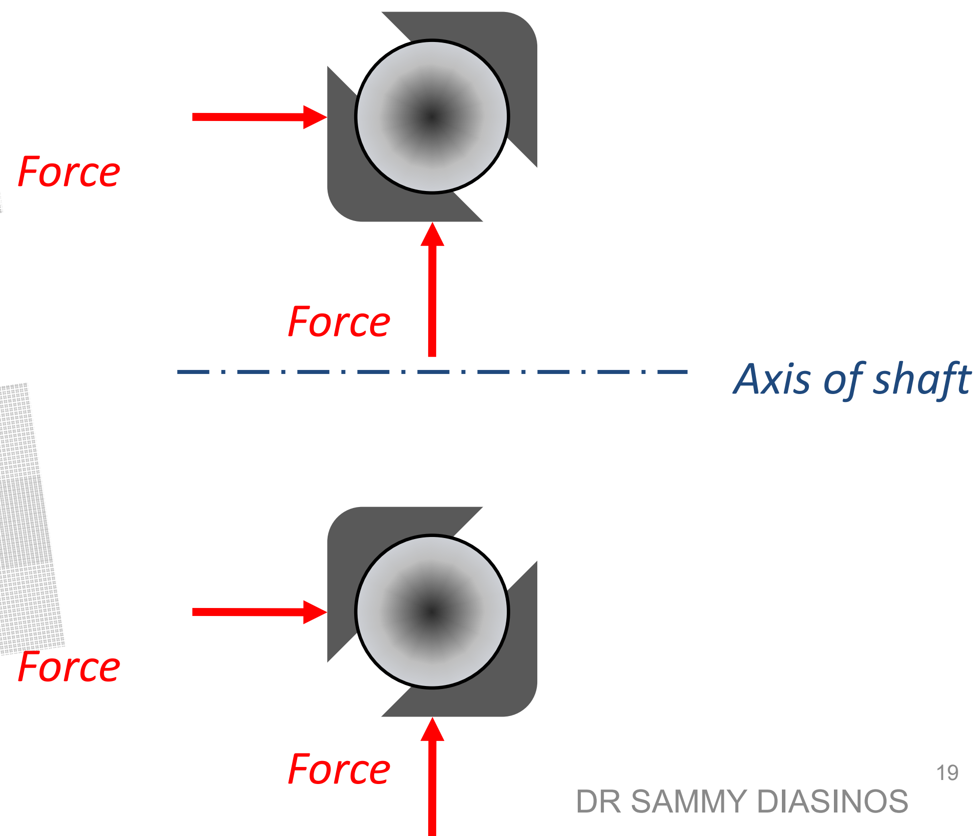
Allowing shafts to rotate - bearings

When a combination of loads both parallel and normal to the shaft are required to be accommodated, tapered or angled bearings are more appropriate with a variety of angles available.



Tapered roller bearing



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Allowing shafts to rotate - bearings

Their are many more types of bearings than what we have discussed today, your role as an engineer is to ensure that you choose the most suitable for your application, all ways check the loads and speeds provided in the catalogues by supplier/manufacturer!



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Allowing shafts to rotate

Example 1: A light remote control car that will be mass produced from plastic and sold for $50 requires either bearings or bushings to support the wheel axles. Which would you choose and why?

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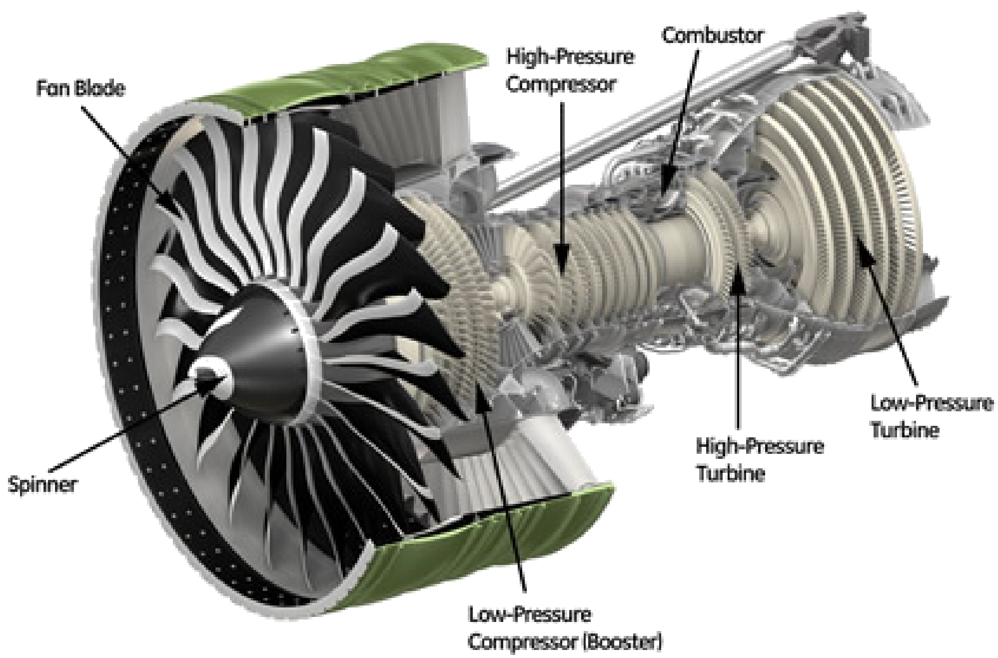
Allowing shafts to rotate

Example 2: A jet engines main shaft can achieve rotation speeds greater than 20000rpm. What components would be most suitable to support the main shaft and why?



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Design Problem – Rotating Shaft System

Who rode a bike to university today?

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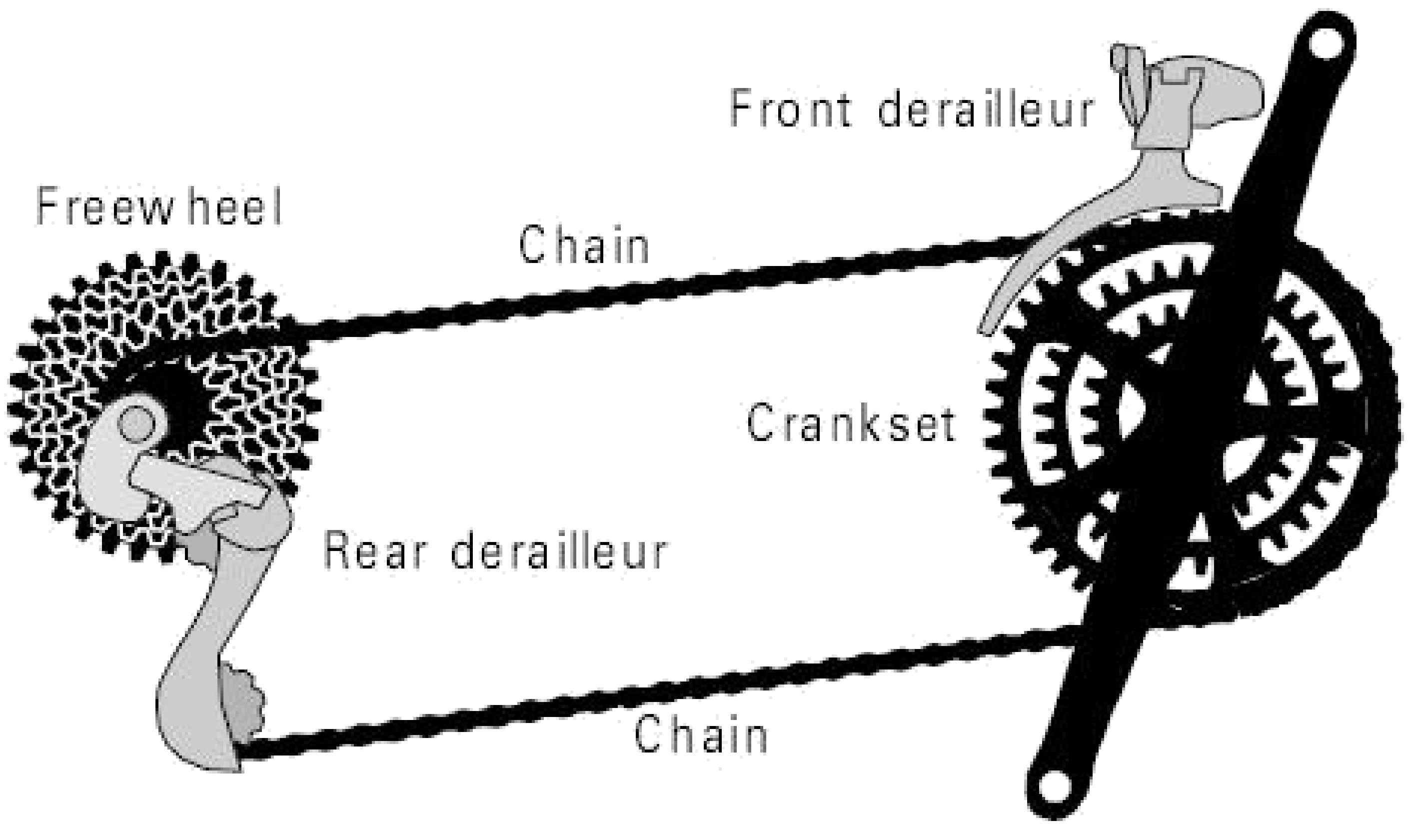
Design Problem – Rotating Shaft System

Why does a bicycle have multiple gears?



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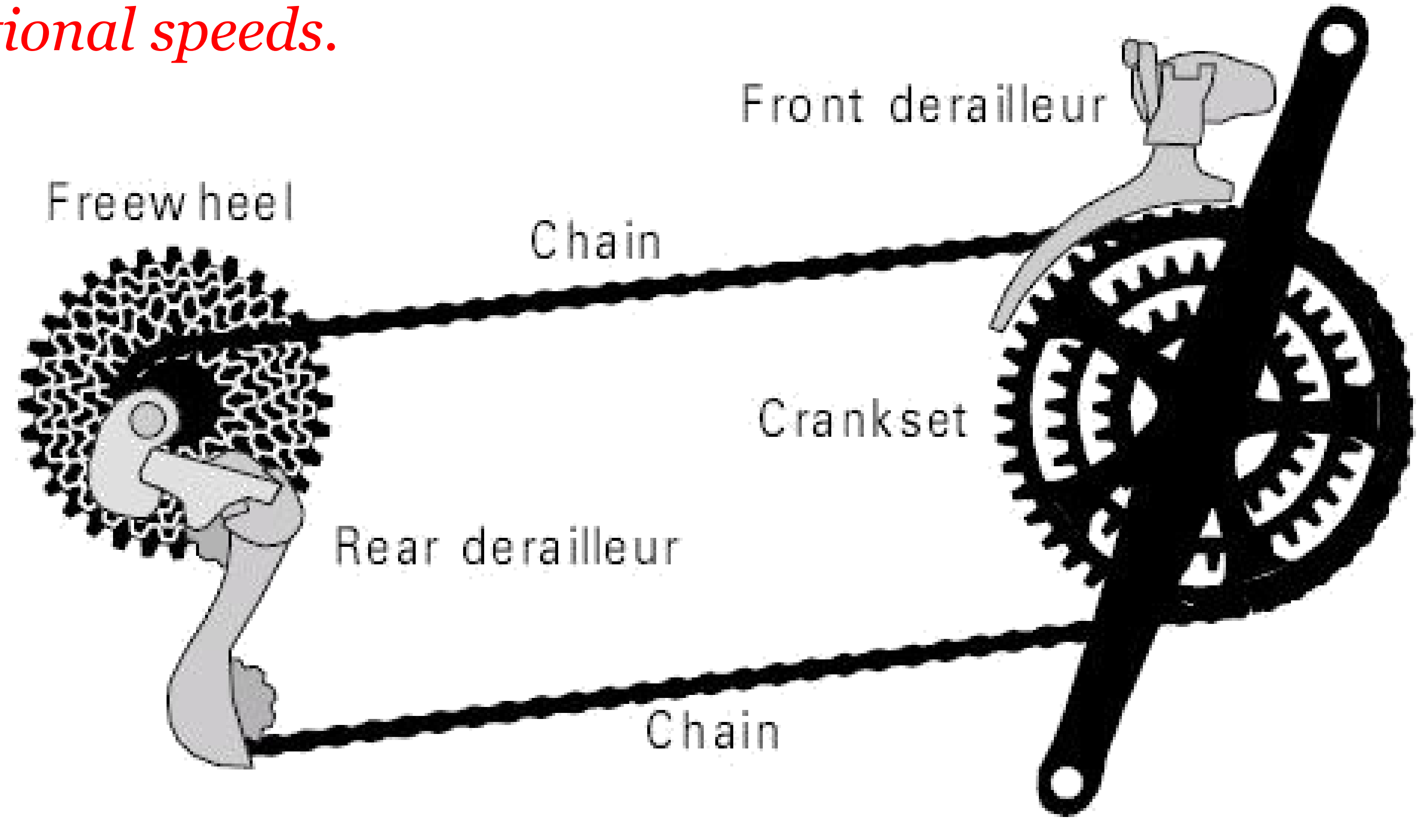
Design Problem – Rotating Shaft System

Why does a bicycle have multiple gears? *They allow for a constant power to be delivered in a range of torques and rotational speeds.*



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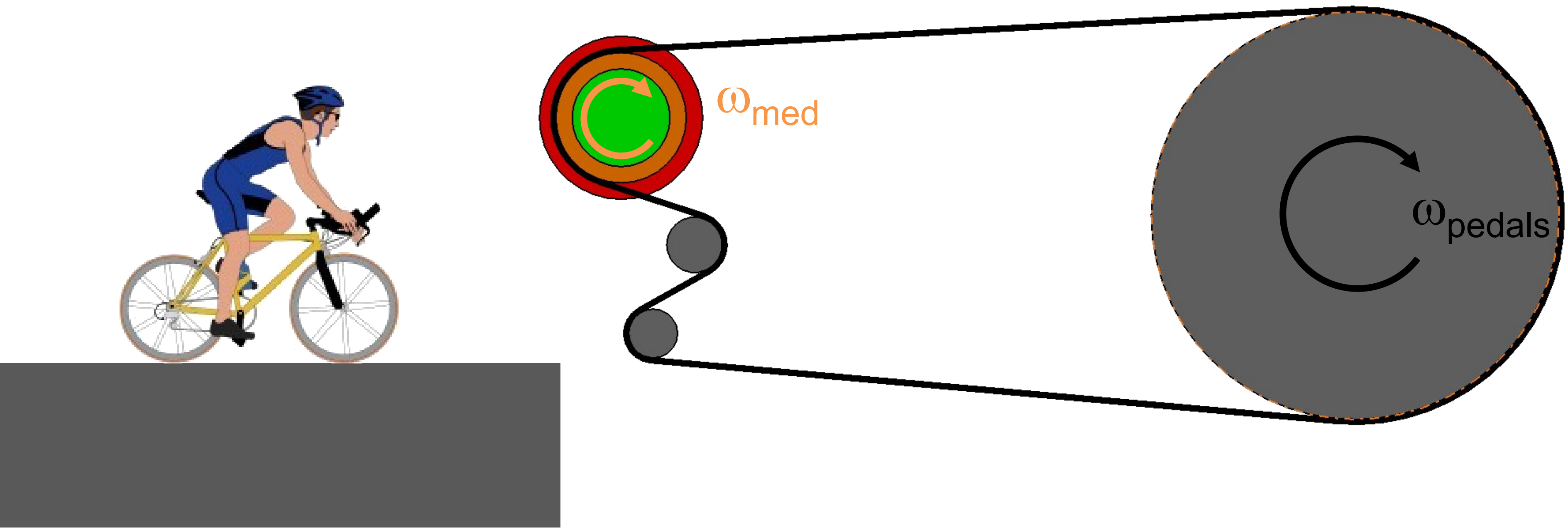
When riding along a horizontal piece of road you may use an intermediate gear to pedal comfortably and achieve a high speed.



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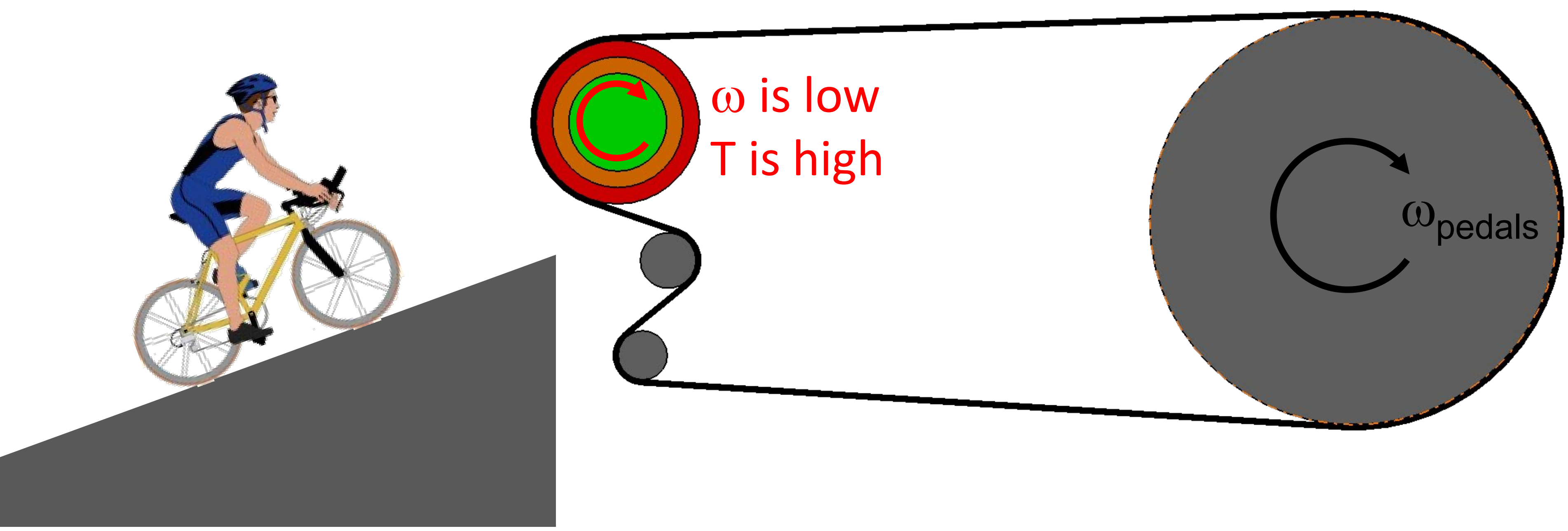
But when the road has an incline, a lower speed gear is used to obtain the required torque to climb comfortably while you compromise your speed.



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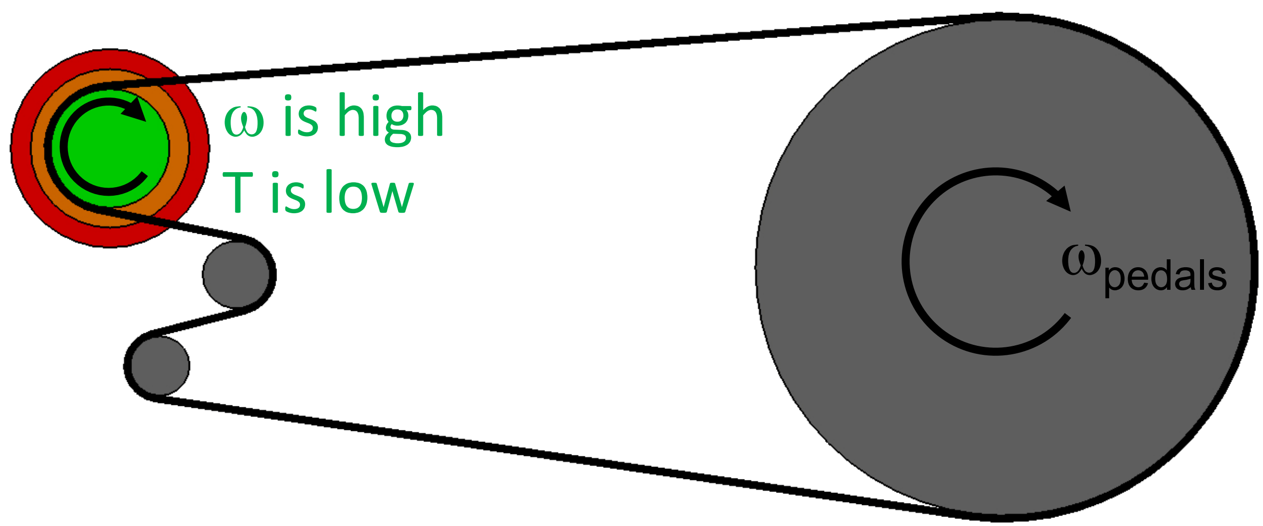
Downhill is very easy to pedal, the load required to accelerate is small so you can use a higher gear to achieve a higher speed without increasing effort.



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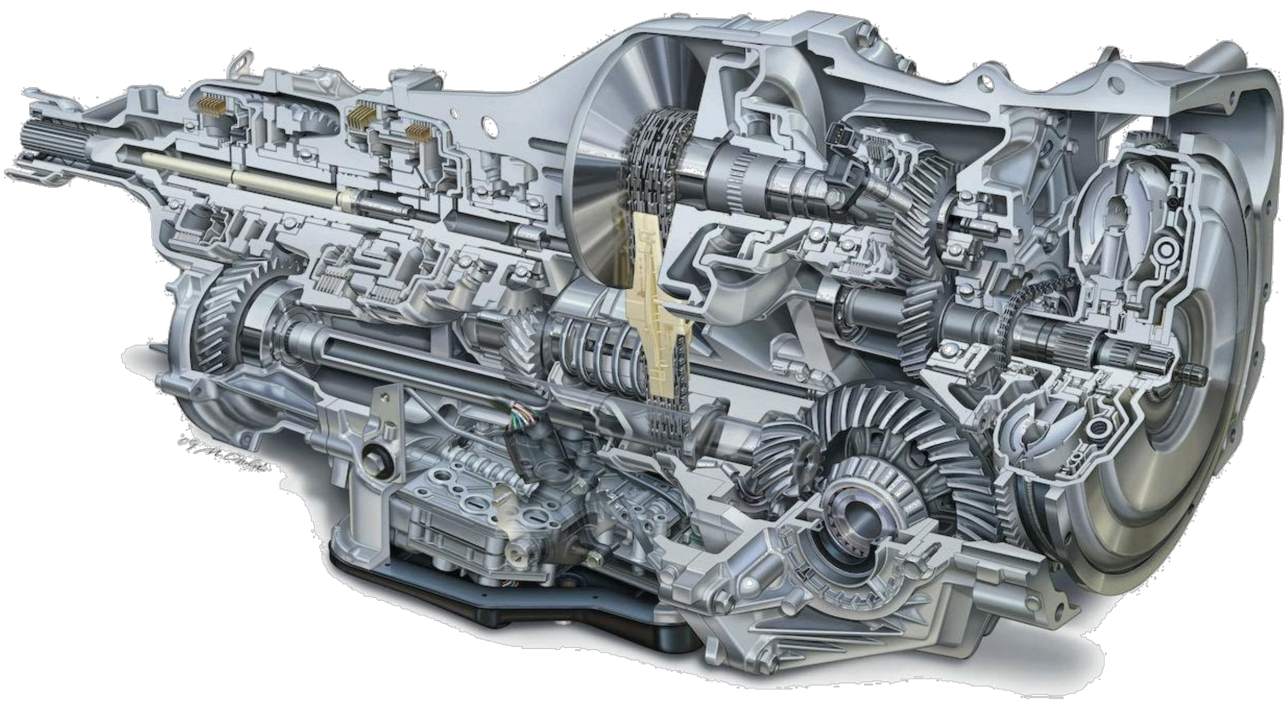
Design Problem – Rotating Shaft System

By changing the gears on a bicycle you compromise the rotational velocity to achieve the torque required, this is identical to the role a car’s gearbox plays.



Design Problem – Rotating Shaft System

So how can we calculate the torque and the angular velocity that we should expect as the output for a rotating system when we have a variation of connections like gears and belts between shafts?

$$Power = Torque \times Angular\ Velocity$$

$$P = T\omega$$

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Angular Velocity Ratio – Gears

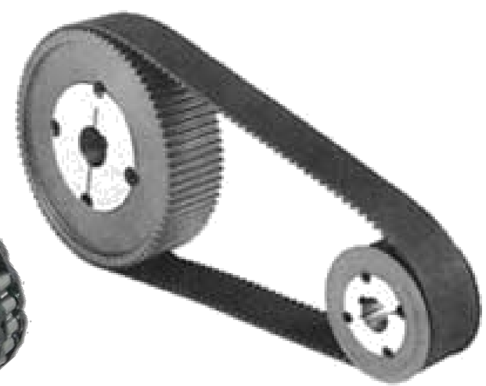
So how can we calculate the torque and the angular velocity that we should expect as the output for a rotating system when we have a variation of connections like gears and belts between shafts?

First we need to determine the relative angular velocity for each shaft which is dependant on the connection…

Gears

Chain drive

Toothed belt



V-belt

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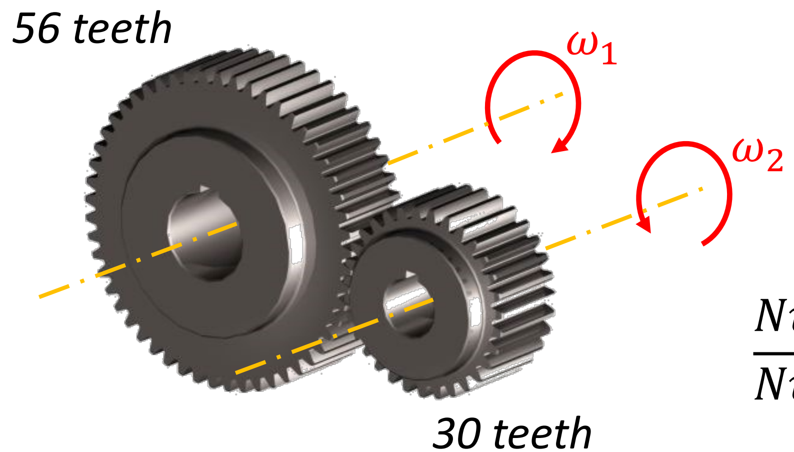
Design Problem – Rotating Shaft System

For gears, the angular velocity ratio is equal to the inverse of the number of teeth between each gear. The direction of rotation is also reversed!



$$\frac{Number\ of\ teeth\ on\ gear\ 1}{Number\ of\ teeth\ on\ gear\ 2} = -\frac{\omega_2}{\omega_1}$$

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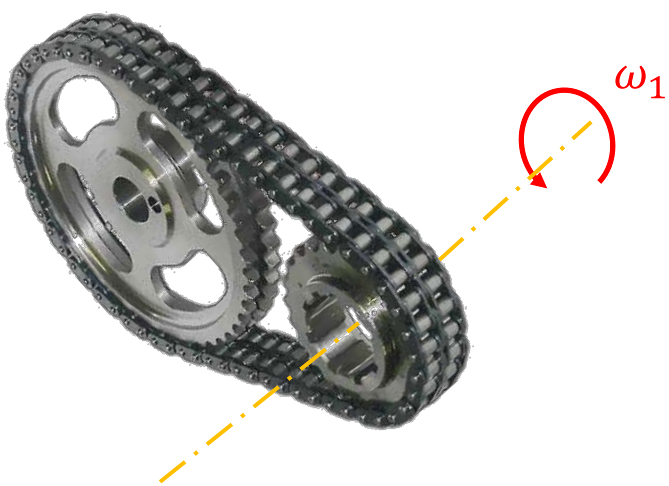
Angular Velocity Ratio – Chain drive

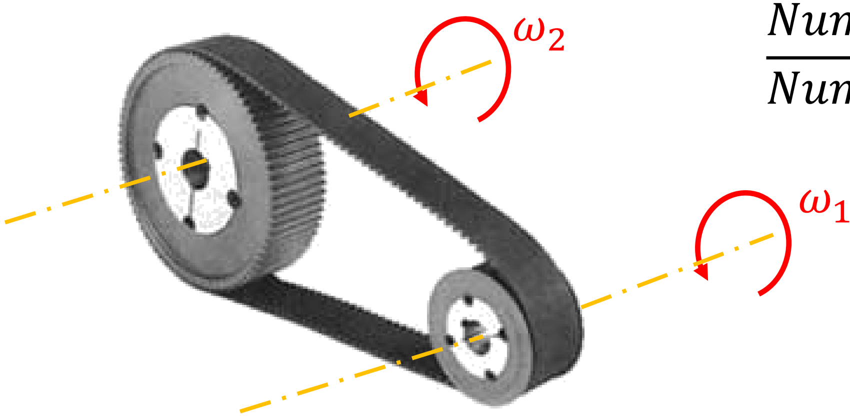
For chain drives, the angular velocity ratio is also equal to the inverse of the number of teeth between each gear but the direction of rotation remains the same!



$$\frac{Number\ of\ teeth\ on\ gear\ 1}{Number\ of\ teeth\ on\ gear\ 2} = \frac{\omega_2}{\omega_1}$$

Angular Velocity Ratio – Timing Belts

For toothed belt drives, the angular velocity ratio is also determined by the number of teeth between each gear and the direction of rotation is not changed unless the belt is crossed.

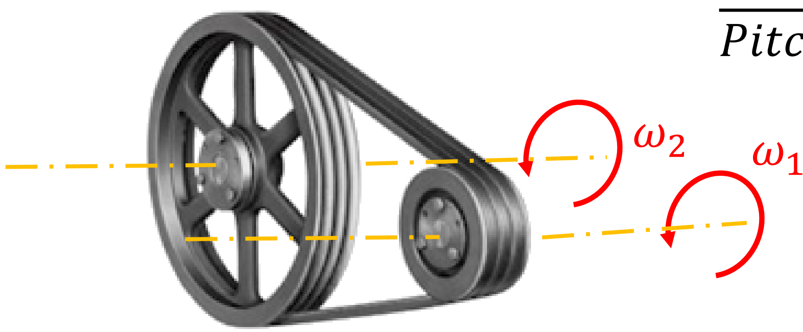


$$\frac{Number\ of\ teeth\ on\ gear\ 1}{Number\ of\ teeth\ on\ gear\ 2} = \frac{\omega_2}{\omega_1}$$

Angular Velocity Ratio – V-Belts

For belt drive systems with no teeth, the angular velocity ratio is equal to the inverse of the two pulley diameters. The rotation direction does not change between the two coupled shafts.

$$\frac{Pitch\ diameter\ of\ pulley\ 1}{Pitch\ diameter\ of\ pulley\ 2} = \frac{\omega_2}{\omega_1}$$



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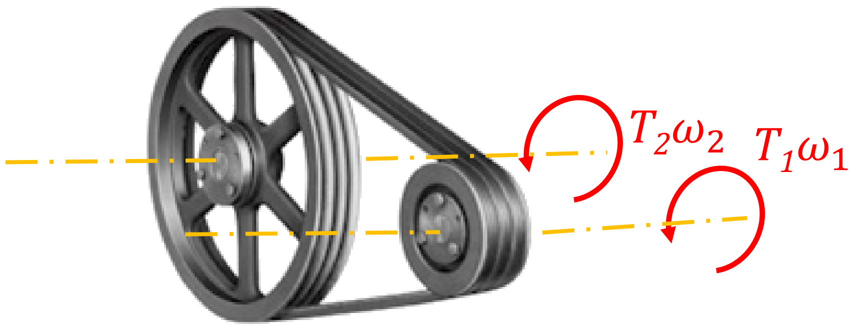
Angular Velocity Ratio – V-Belts

Once the angular velocity has been determined, it is possible to calculate the torque that each shaft will experience. Assuming their are no transmission losses, the power that both shafts have is equal.



$$Power = T\omega$$

$$T_1\omega_1 = T_2\omega_2$$

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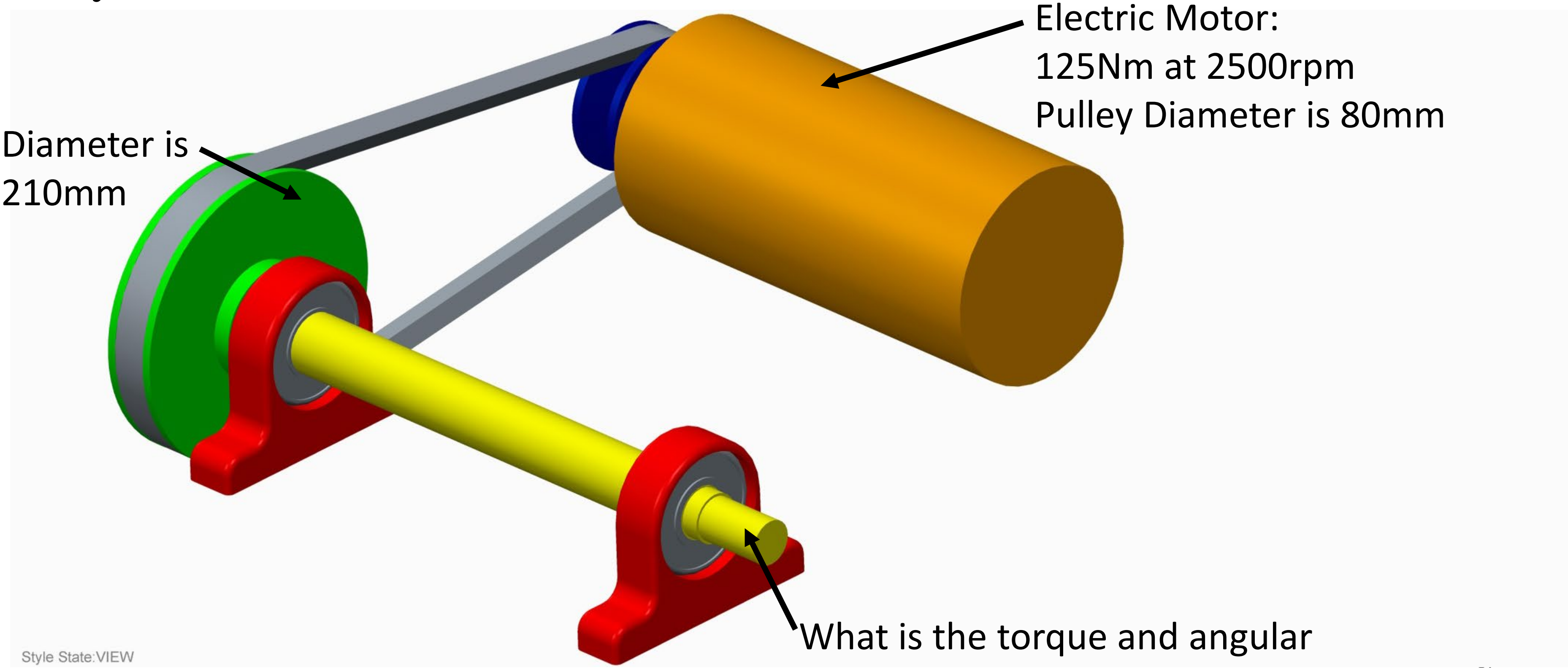
Angular Velocity Ratio

Example 3: Determine the torque and angular velocity at the shaft assuming that their are no transmission losses in the system.



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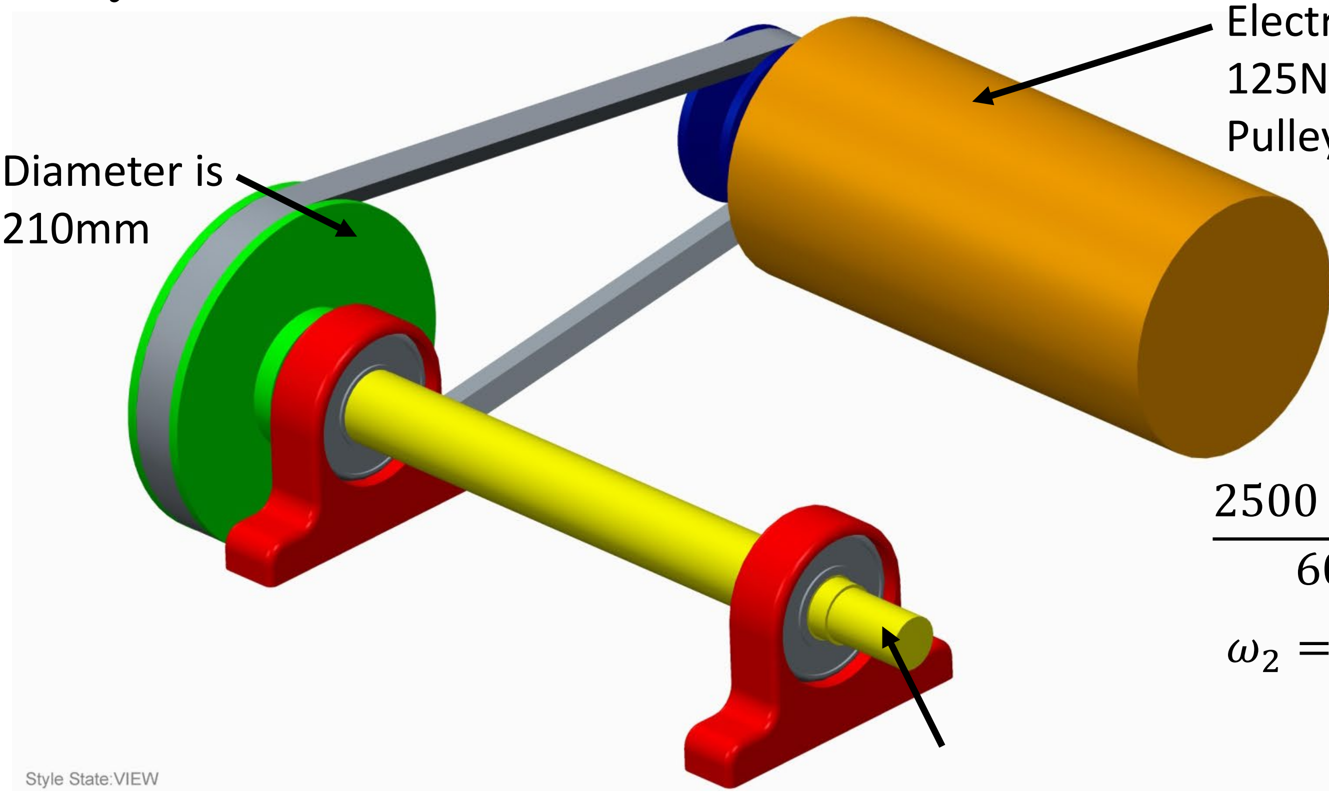
Angular Velocity Ratio

Example 3: Determine the torque and angular velocity at the shaft assuming that their are no transmission losses in the system.



$$T_1\omega_1 = T_2\omega_2$$

$$125 \times \frac{2500 \times 2\pi}{60} = T_2\omega_2$$

$$\omega_1 d_1 = \omega_2 d_2$$

$$\frac{2500 \times 2\pi}{60} \times 0.08 = \omega_2 \times 0.21$$

$$\omega_2 = 952.4\, rpm = 99.73 \frac{rad}{s}$$

$$\therefore T_2 = 328.1\, Nm$$

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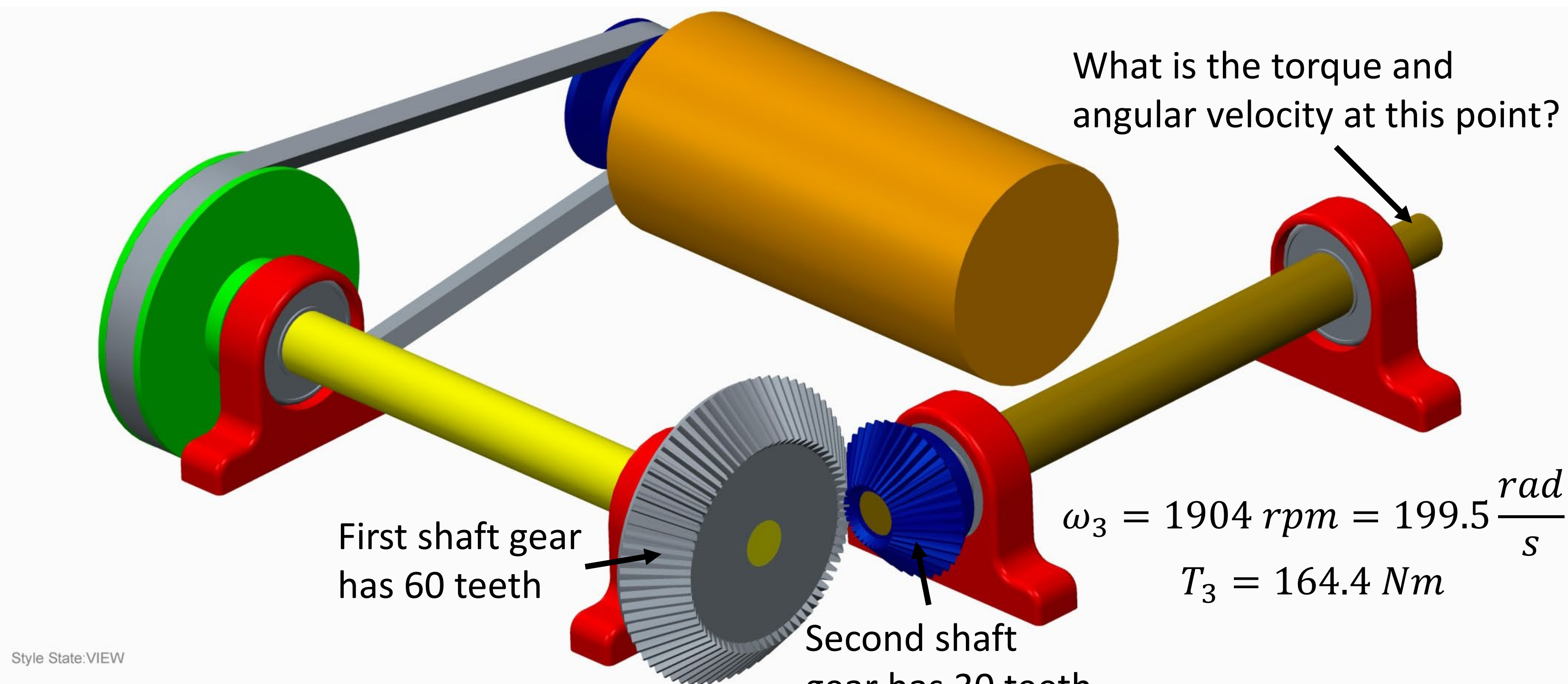
Angular Velocity Ratio

Example 4: Determine the torque and angular velocity at the final shaft assuming that their are no transmission losses in the system.



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Angular Velocity Ratio

Example 5: While riding my bicycle along a flat and horizontal road, I can achieve a constant speed of 35km/h while using the highest gear. If I must change down gears to ride up the incline without exerting additional effort, how fast will I be travelling up the incline?

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Angular Velocity Ratio

Example 5: While riding my bicycle along a flat and horizontal road, I can achieve a constant speed of 35km/h while using the highest gear. If I must change down gears to ride up the incline without exerting additional effort, how fast will I be travelling up the incline?

$$v = \omega r$$

$$\omega_1 = \frac{35 \times \frac{1000}{3600}}{0.035}$$

$$\omega_1 = 27.78\frac{rad}{s}$$

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Angular Velocity Ratio

Example 5: While riding my bicycle along a flat and horizontal road, I can achieve a constant speed of 35km/h while using the highest gear. If I must change down gears to ride up the incline without exerting additional effort, how fast will I be travelling up the incline?

$$v = \omega r$$

$$\omega_2 = 27.78 \times \frac{4}{5}$$

$$\omega_2 = 22.22 \frac{rad}{s}$$

$$\therefore v_2 = 7.7778 \frac{m}{s}$$

$$\approx 28 \, {}^{km}/_{h}$$

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Coupling selection

When would we use each type of connection, it could be determined by the torque that is required to be transmitted:

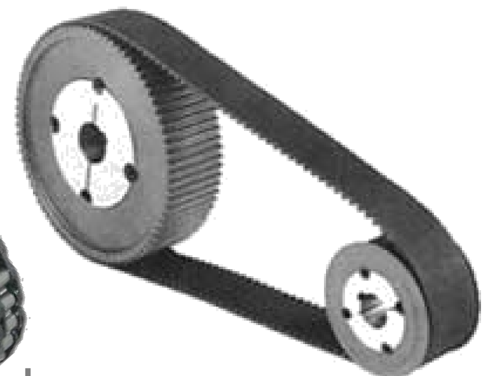
1 Gears

Highest torque transfer capability

2 Chain drive

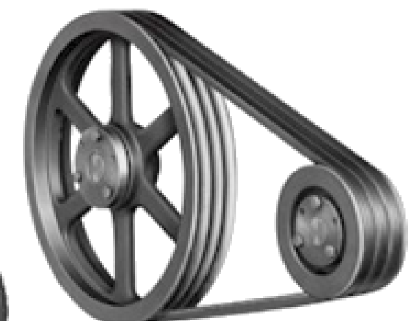
High torque transfer capability

3 Toothed belt



Less than chain, but does not need as high belt tensions to achieve V-belt torque equivalent

4 V-belt



Lowest torque capability, dependant on the tension in the belts

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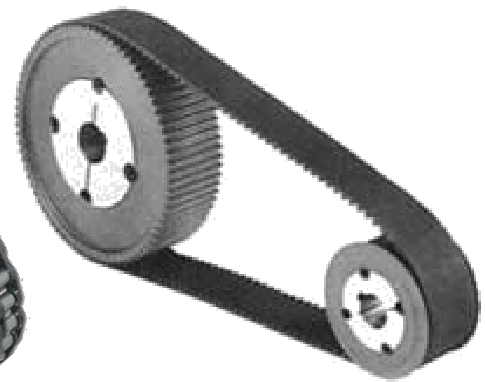
Coupling selection

When would we use each type of connection, it could be determined by the relative position of shafts:

Gears

Chain drive

Toothed belt



V-belt

Shafts are close together or crossing

Shafts can be some distance apart, but must be parallel, some misalignment may be tolerated

Large misalignments can be tolerated

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Coupling selection

When would we use each type of connection, it could be determined by the required efficiency:

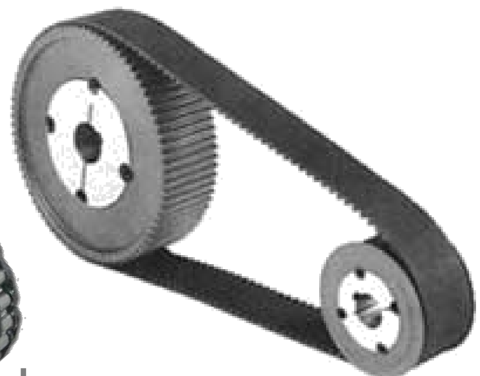
3 Gears

For ratio's of 1:1 to 6:1 94% to 98% efficient

1 Chain drive

Up to 98% efficient when well lubricated

1 Toothed belt



Up to 98% efficient depending on tension

4 V-belt

88% to 95% efficient depending on the contact angle and tension

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Coupling selection

When would we use each type of connection, it could be determined by which is most cost effective:

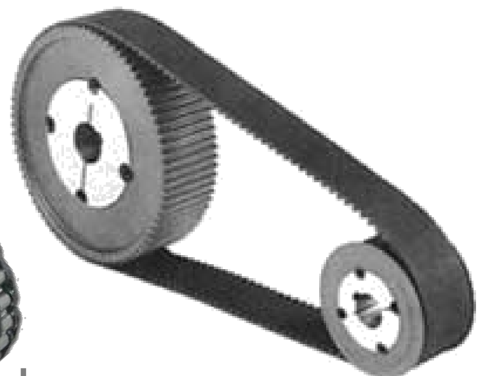
1 or 4 Gears

Very dependent on material selected.

2 Chain drive

Many available gears and chains but selection is limited to pitch of chain

3 Toothed belt



Pulley selection is limited due to pitch of teeth selected for belt

1 V-belt

Many available pulleys and belts making them very cost effective

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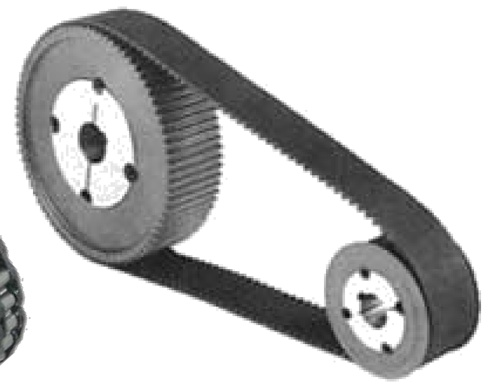
Coupling selection

When would we use each type of connection, it could be determined by the angular velocity ratio that needs to be achieved:

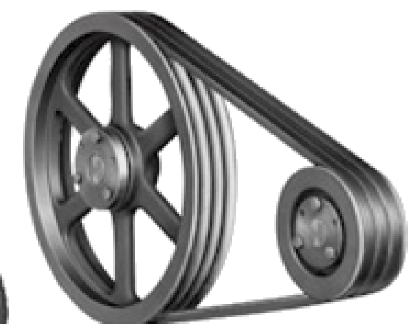
1 Gears

2 Chain drive

3 Toothed belt



4 V-belt



Very large ratio changes possible

Ratio changes are limited by the contact angle of the belt/chain around the pulley/gears and is also affected by the distance between the two rotating shafts. Efficiency compromised as ratio increases.

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Coupling selection

When would we use each type of connection, it could be determined by the noise level required:

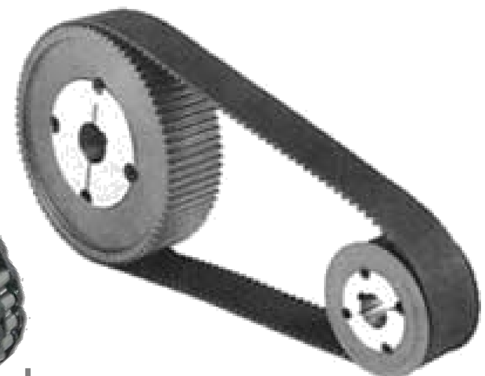
2 Gears

Helical gears can reduce noise further in comparison.

3 Chain drive

Noise dependant on level of lubrication

1 Toothed belt



Quietest option of all those listed here

4 V-belt

Noisy particularly when belt tension is low and return portion achieves harmonic

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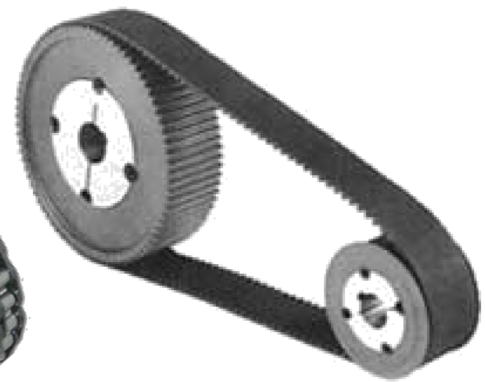
Coupling selection

When would we use each type of connection, it could be determined by the accuracy that relative position must be achieved:

1 Gears

1 Chain drive

1 Toothed belt



4 V-belt

No slip possible unless failure occurs making them very suitable for applications where the relative position of the shafts is crucial.

Belts can slip when the tension is not sufficient.

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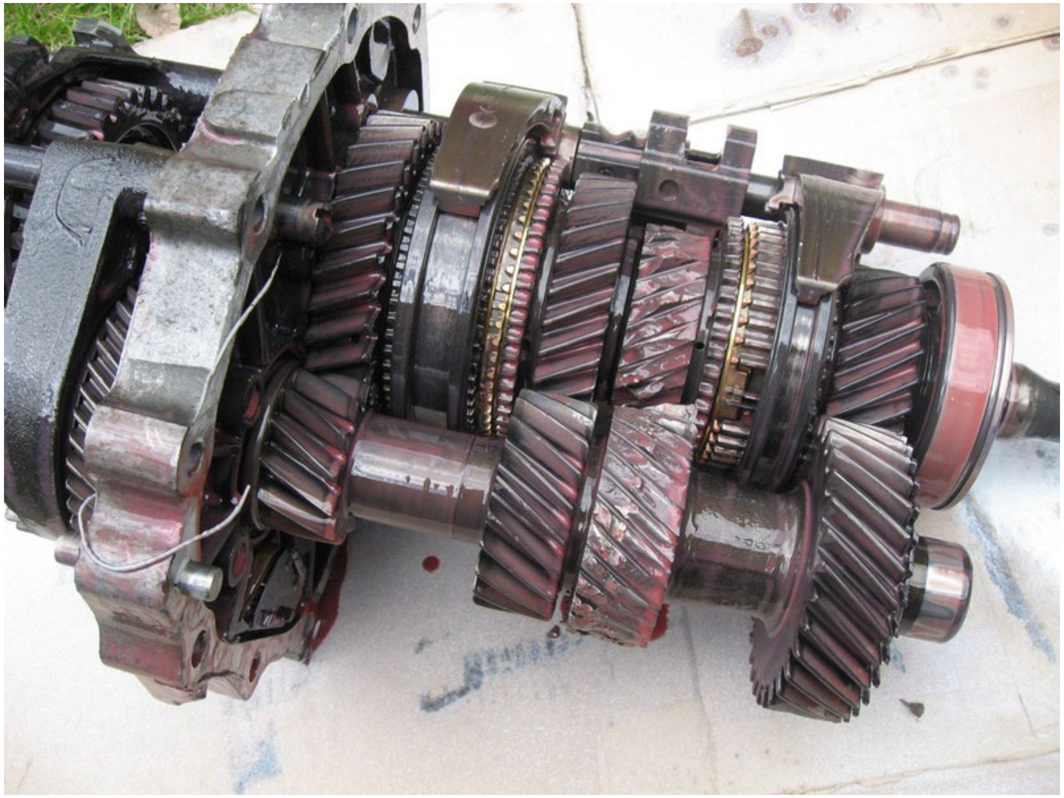
Coupling selection

Application of Gears - *Differentials*

Application of Gears - *Manual and Semi-Automatic Gearboxes*



Application of Chains - *Motorbikes*

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Coupling selection

Application of Chains – *Chainsaws*

Coupling selection

Application of Timing Belts *– driving cams in engines*

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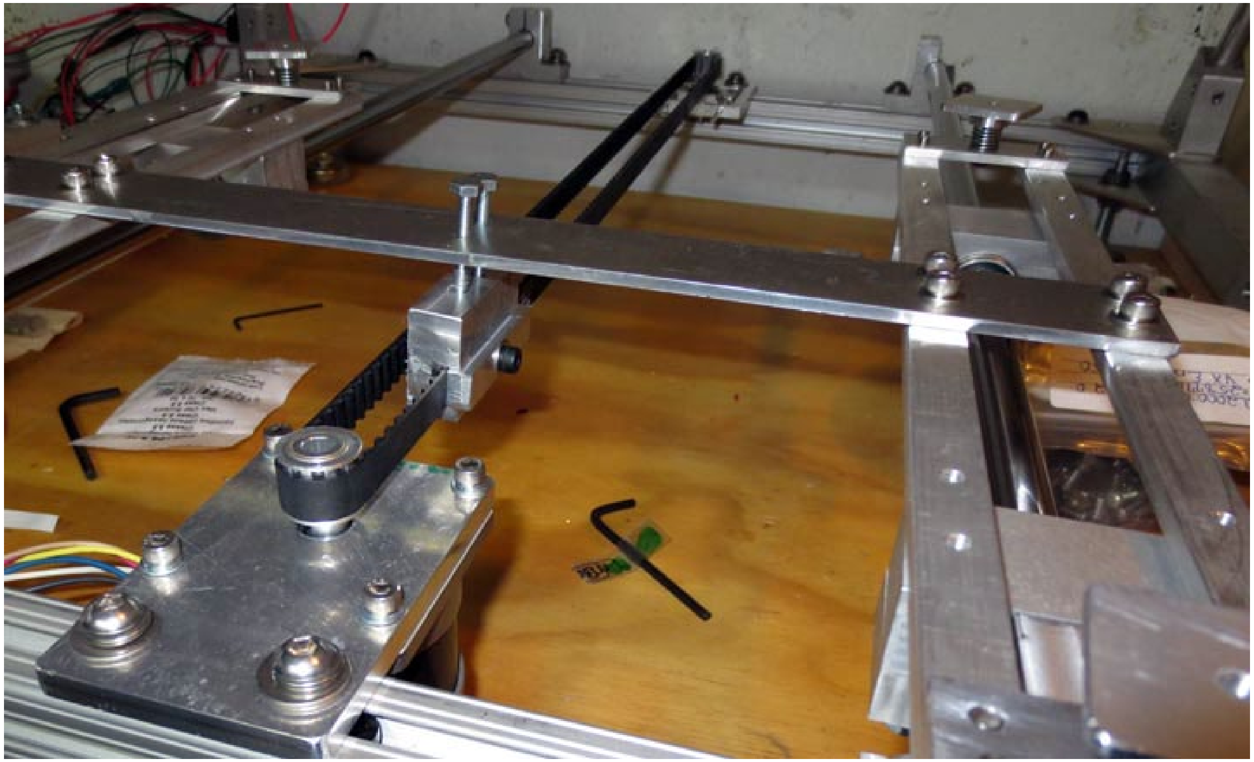
Coupling selection

Application of Timing Belts *– Traverse (3d printer)*



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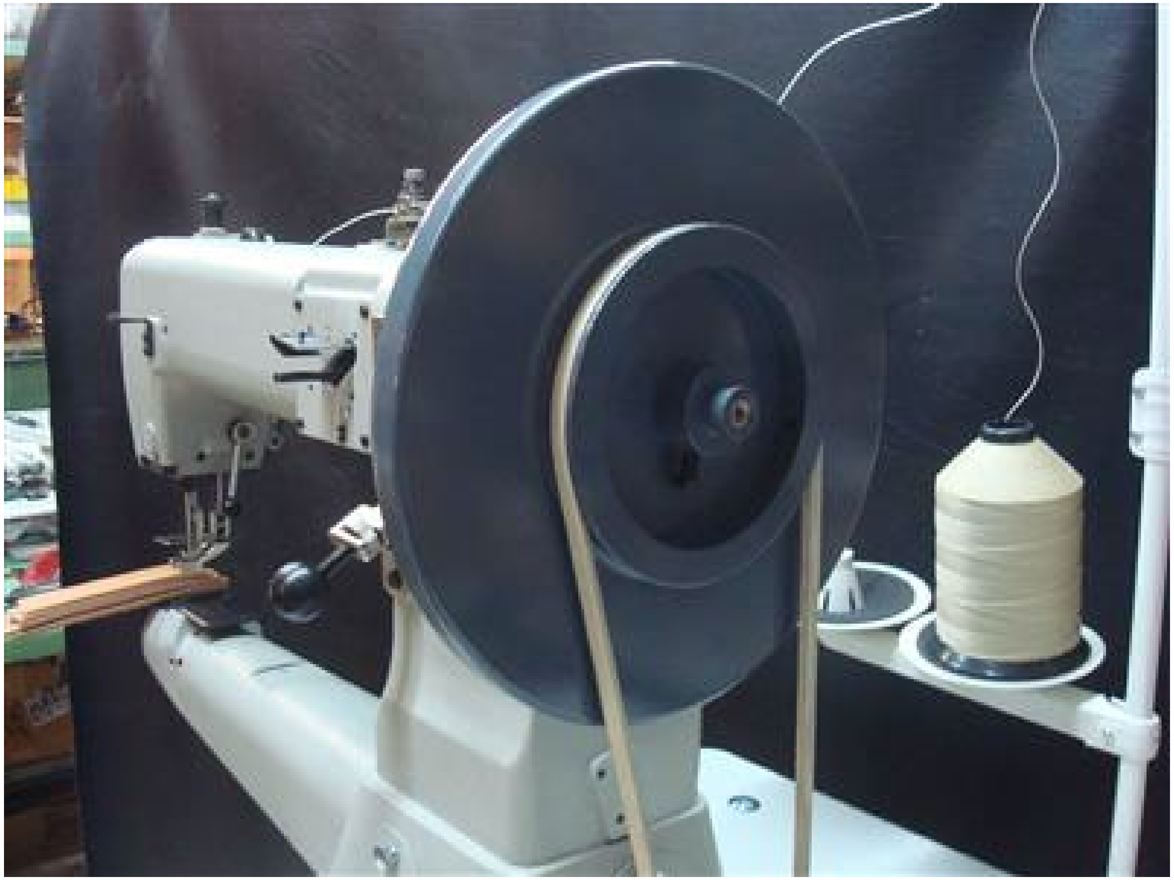
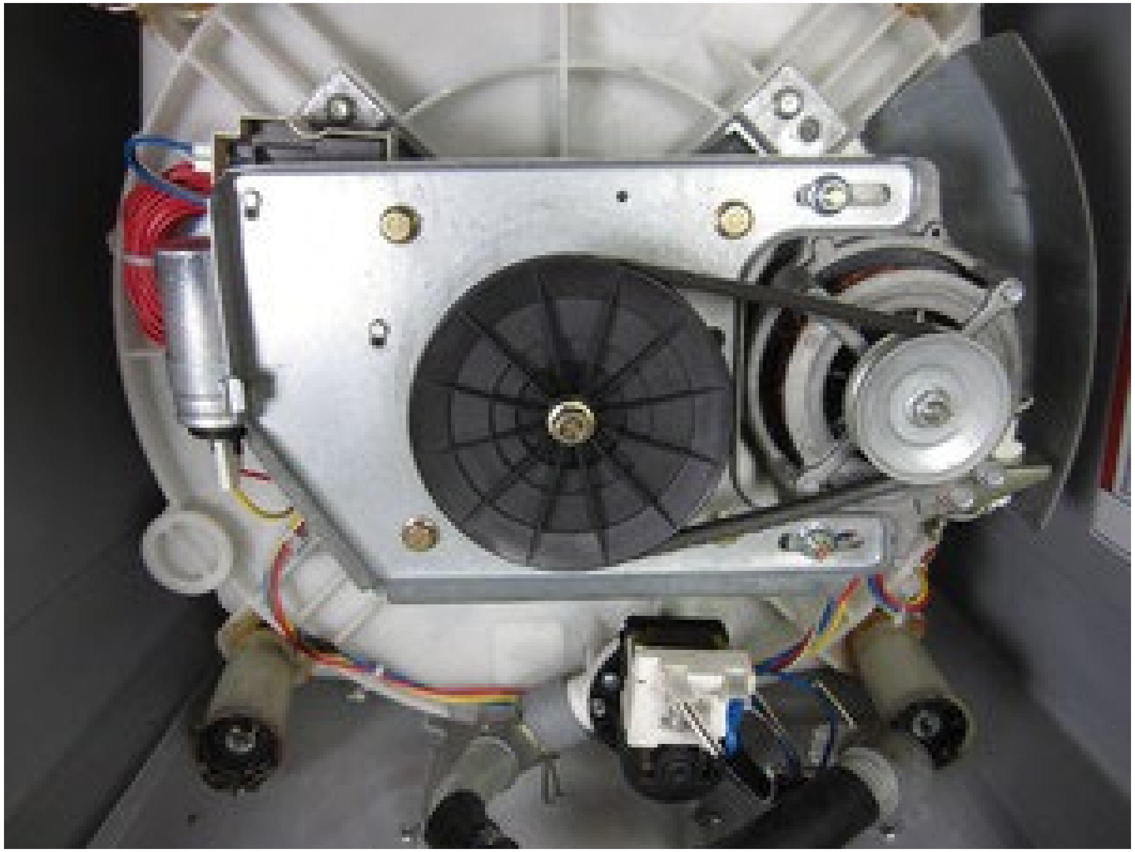
Coupling selection

Application of V-Belts – *Sewing machines*



Coupling selection

Application of V-Belts – *washing machines*



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Coupling selection

Application of Flat Belts *– non-parallel shafts*

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Losses in the system

While the analysis conducted thus far for the power transferred between shafts assumes that their is no power loss, this is not necessarily all ways an accurate assumption to make.

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Losses in the system

It has all ready been identified that different connecting methods between shafts are not perfectly efficient.

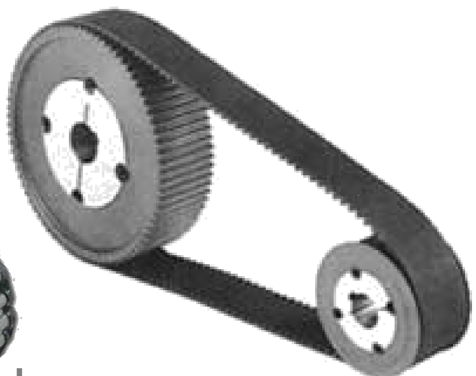
3 Gears

For ratio's of 1:1 to 6:1 94% to 98% efficient

1 Chain drive

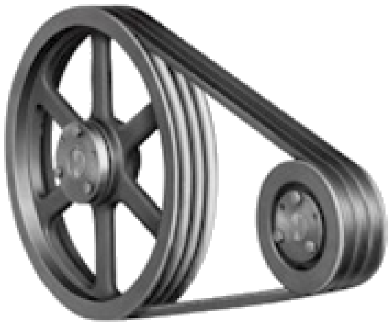
Up to 98% efficient when well lubricated

1 Toothed belt



Up to 98% efficient depending on tension

4 V-belt



88% to 95% efficient depending on the contact angle and tension

Power has also been identified to be the multiple of angular velocity and torque, so given that most coupling techniques have fixed angular velocity ratio's (there is no slip), inefficiencies result in a loss of torque.

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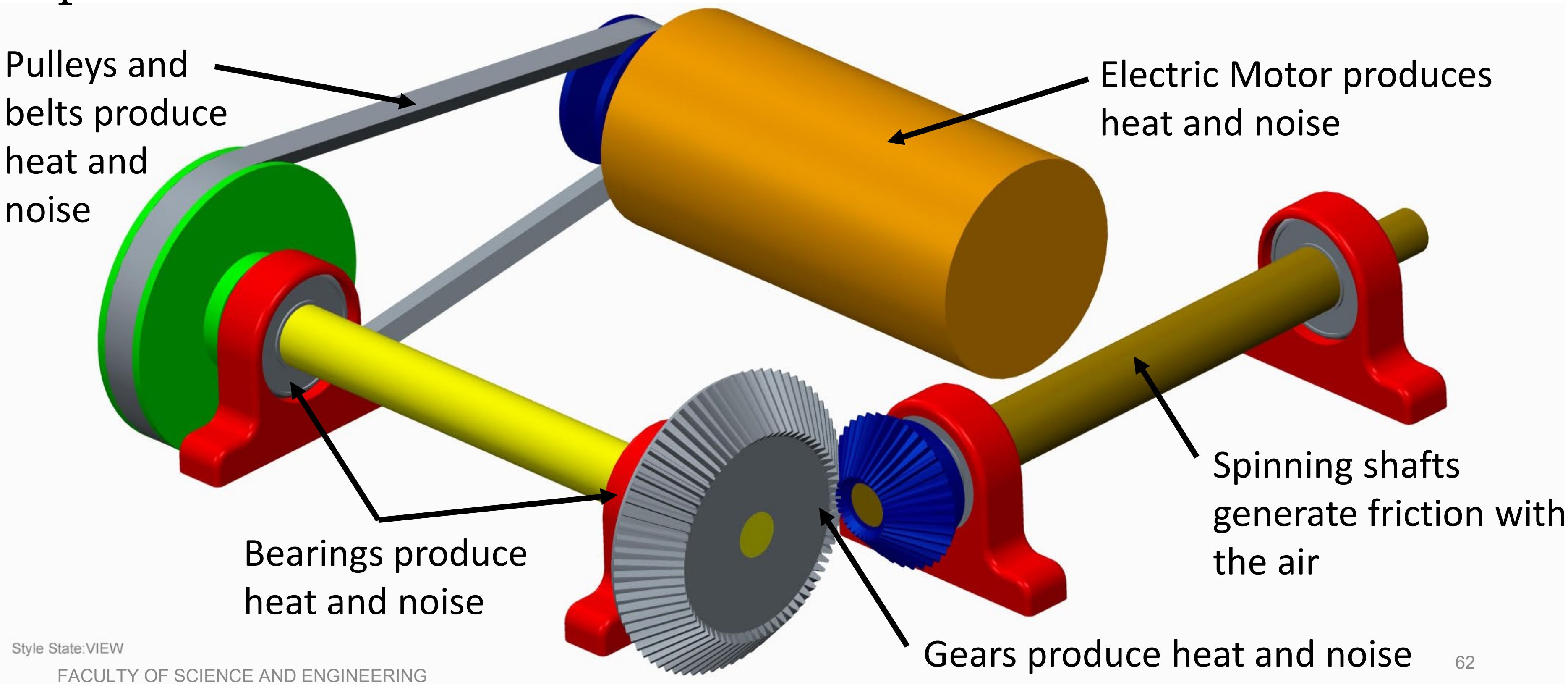
Anywhere where mechanical power may be converted to heat or any other non mechanical form, their is the potential of having power losses.



Losses in the system

Even bearings that drastically reduce the friction in comparison to a bushing still have some measureable inefficiencies related to their operating load and speed which can be estimated using the following equation:

$$M = 0.5\mu P d$$

Where:

M = frictional moment (Nm)

μ = constant coefficient of friction

P = equivalent dynamic bearing load (N)

d = bearing bore diameter (m)

http://www.skf.com/group/products/bearings-units-housings/ball-bearings/principles/friction/estimating-frictional-moment/index.html

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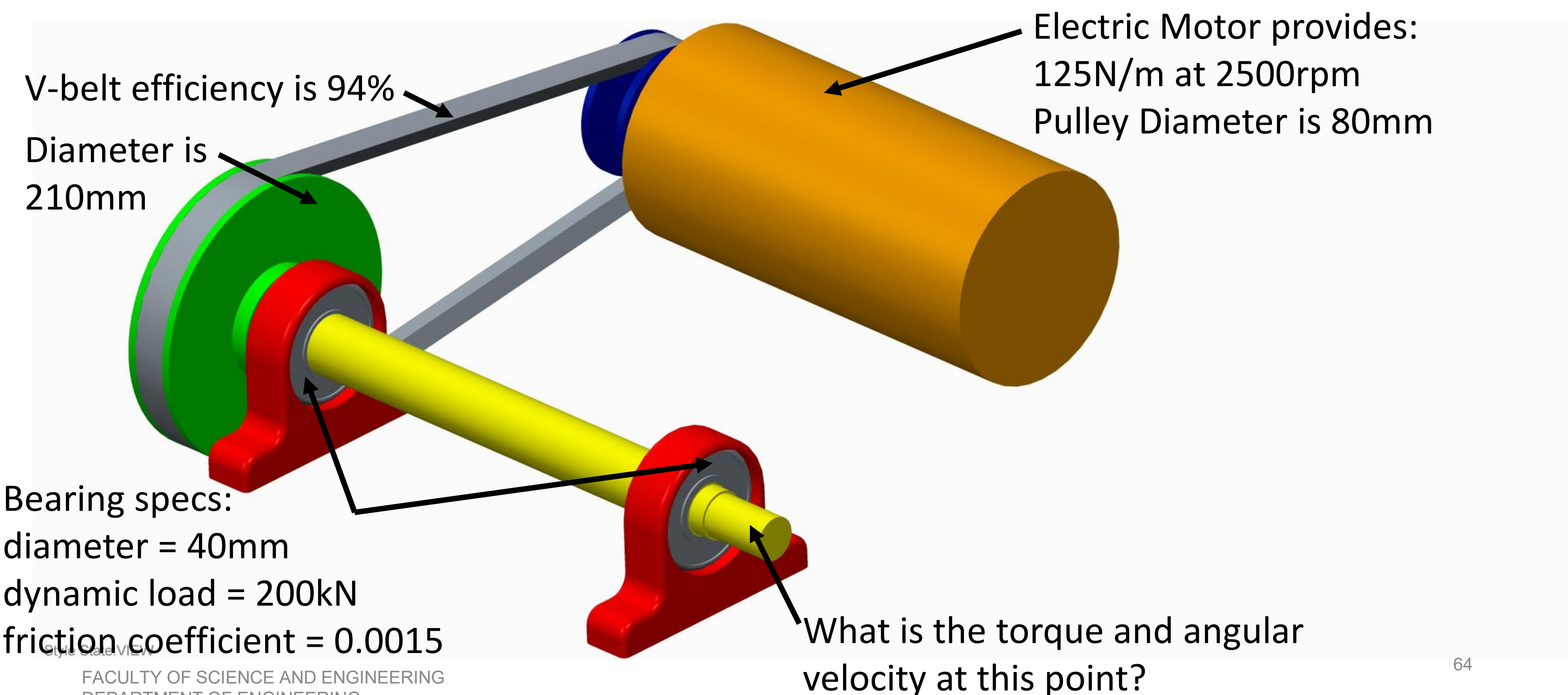
Example 6: Determine the torque and angular velocity of the shaft considering the transmission losses in the system.



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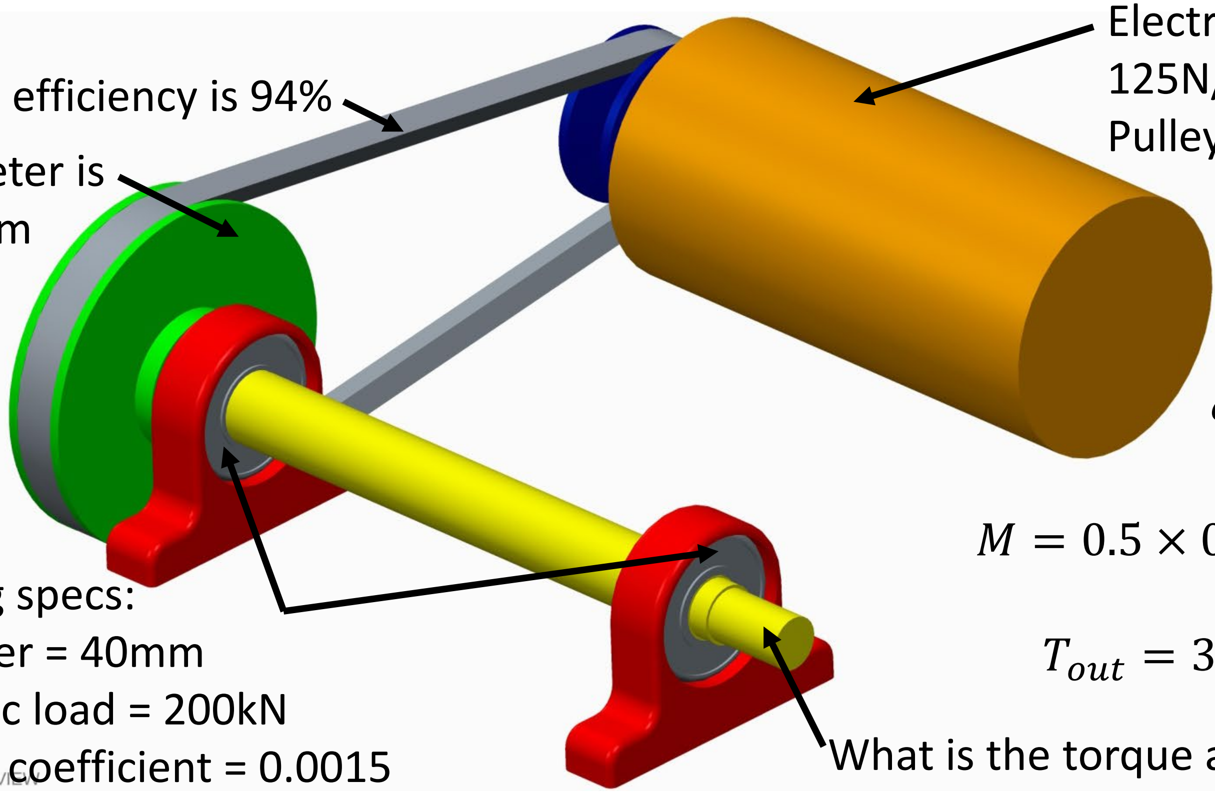
Losses in the system

Example 6: Determine the torque and angular velocity of the shaft considering the transmission losses in the system.



$$T_2 = 328.1\ Nm$$

$$\omega_2 = 99.73\,\frac{rad}{s}$$

$$M = 0.5 \times 0.0015 \times 200 \times 10^3 \times 0.04$$

$$M = 6\ Nm$$

$$T_{out} = 328.1 - 2M = 296.5\ Nm$$

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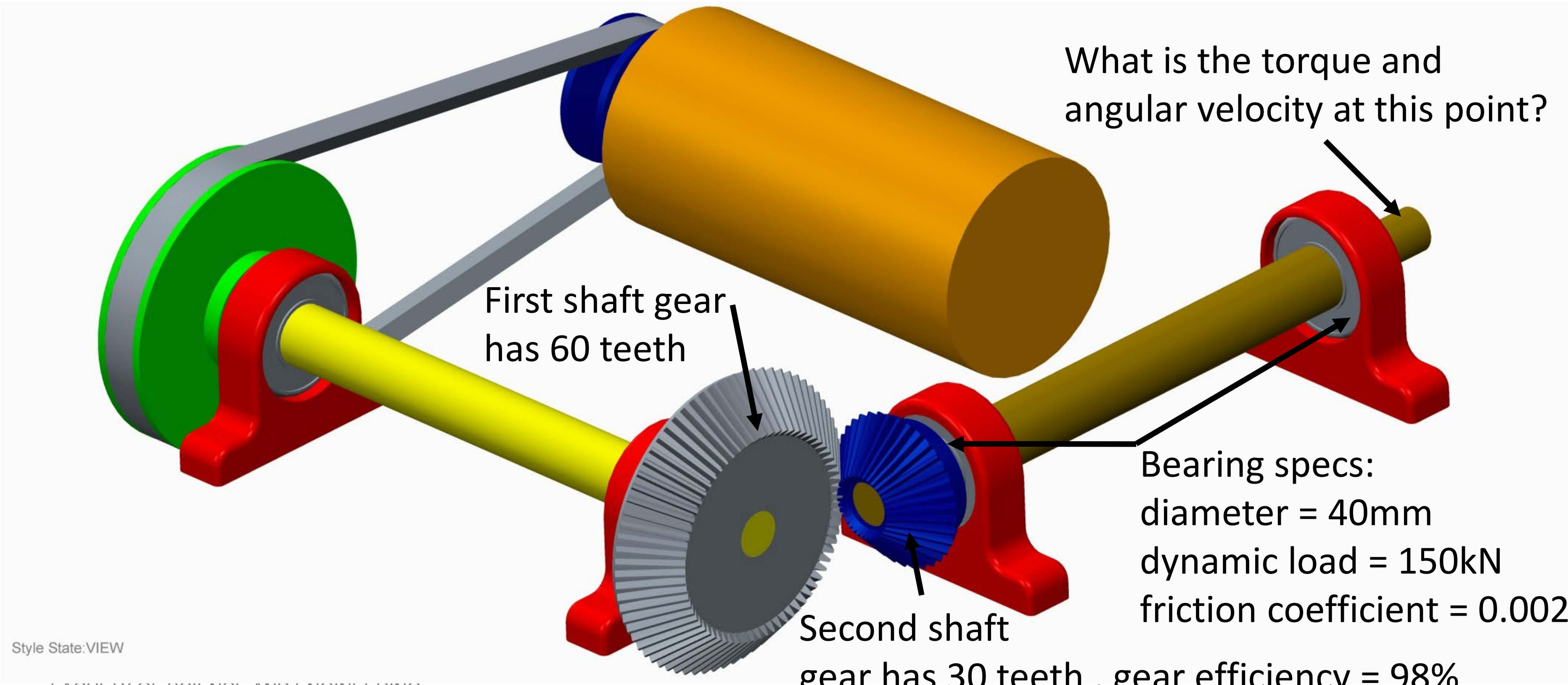
Losses in the system

Example 7: Determine the torque and angular velocity of the final shaft considering the transmission losses in the system.



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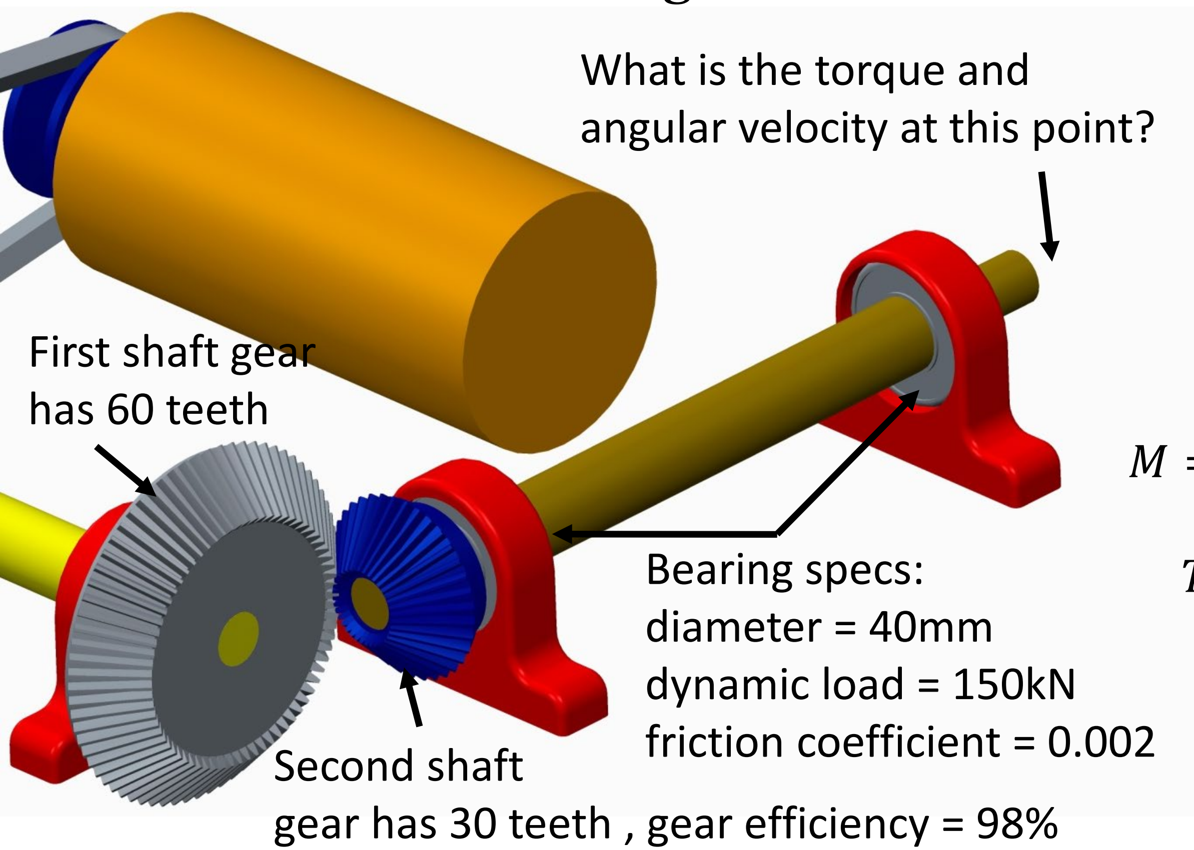
Losses in the system

Example 7: Determine the torque and angular velocity of the final shaft considering the transmission losses in the system.



$$T_{60T} = 296.5\ Nm$$

$$\omega_2 = 99.73\frac{rad}{s}$$

$$T_{30T} = 296.5 \times \frac{30}{60} \times 0.98$$

$$T_{30T} = 145.3\ Nm$$

$$M = 0.5 \times 0.002 \times 150 \times 10^3 \times 0.04$$

$$M = 6\ Nm$$

$$T_{out} = 145.3 - 2M = 133.3\ Nm$$

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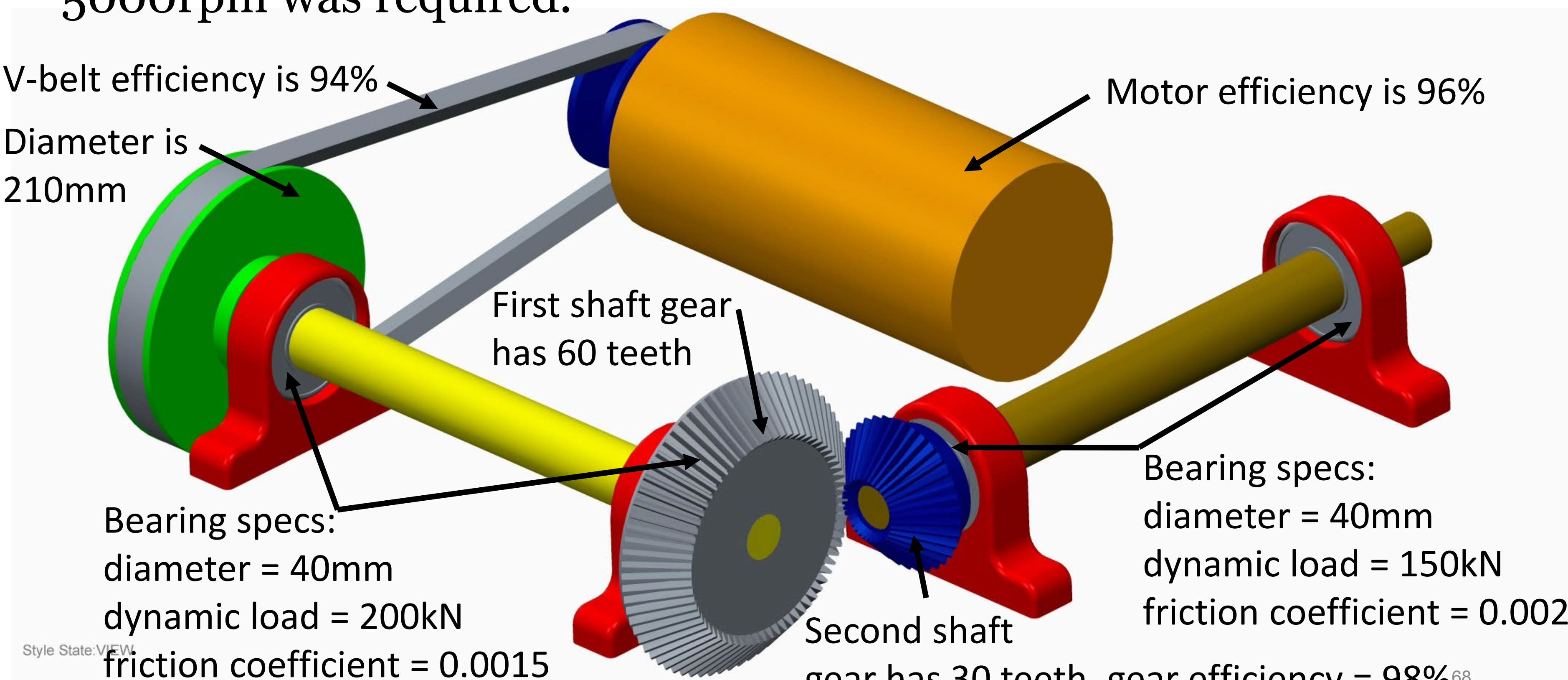
Example 8: Determine the power consumed by the motor if at the final shaft a torque of 100N/m at an angular velocity of 5000rpm was required.



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Losses in the system

Example 8: Determine the power consumed by the motor if at the final shaft a torque of 100N/m at an angular velocity of 5000rpm was required.

$$\omega_{out} = \omega_{in} \times \frac{d_1}{d_2} \times \frac{N_1}{N_2}$$

$$5000 = \omega_{in} \times \frac{80}{210} \times \frac{60}{30}$$

$$\omega_{in} = 6562.5\ rpm = 687.2 \frac{rad}{s}$$

$$M = 6\ Nm$$

For each bearing as per previous examples

$$P_{out} = \eta P_{in}$$

$$P = T\omega$$

$$T_{in} = \frac{[T_{out} + 2M] \times \omega_{out}}{\eta \times \omega_{in}}$$

$$T_{in,gears} = \frac{[100 + 2 \times 6] \times 5000 \times \frac{2\pi}{60}}{0.98 \times 2500 \times \frac{2\pi}{60}}$$

$$T_{in,gears} = 228.57\ Nm$$

$$T_{in,motor} = \frac{[228.57 + 2 \times 6] \times 2500 \times \frac{2\pi}{60}}{0.94 \times 2500 \times \frac{2\pi}{60} \times \frac{210}{80}}$$

$$T_{in,motor} = 97.495$$

$$\therefore P_{motor} = 66{,}997.6\ W \approx\ 67\ kW$$

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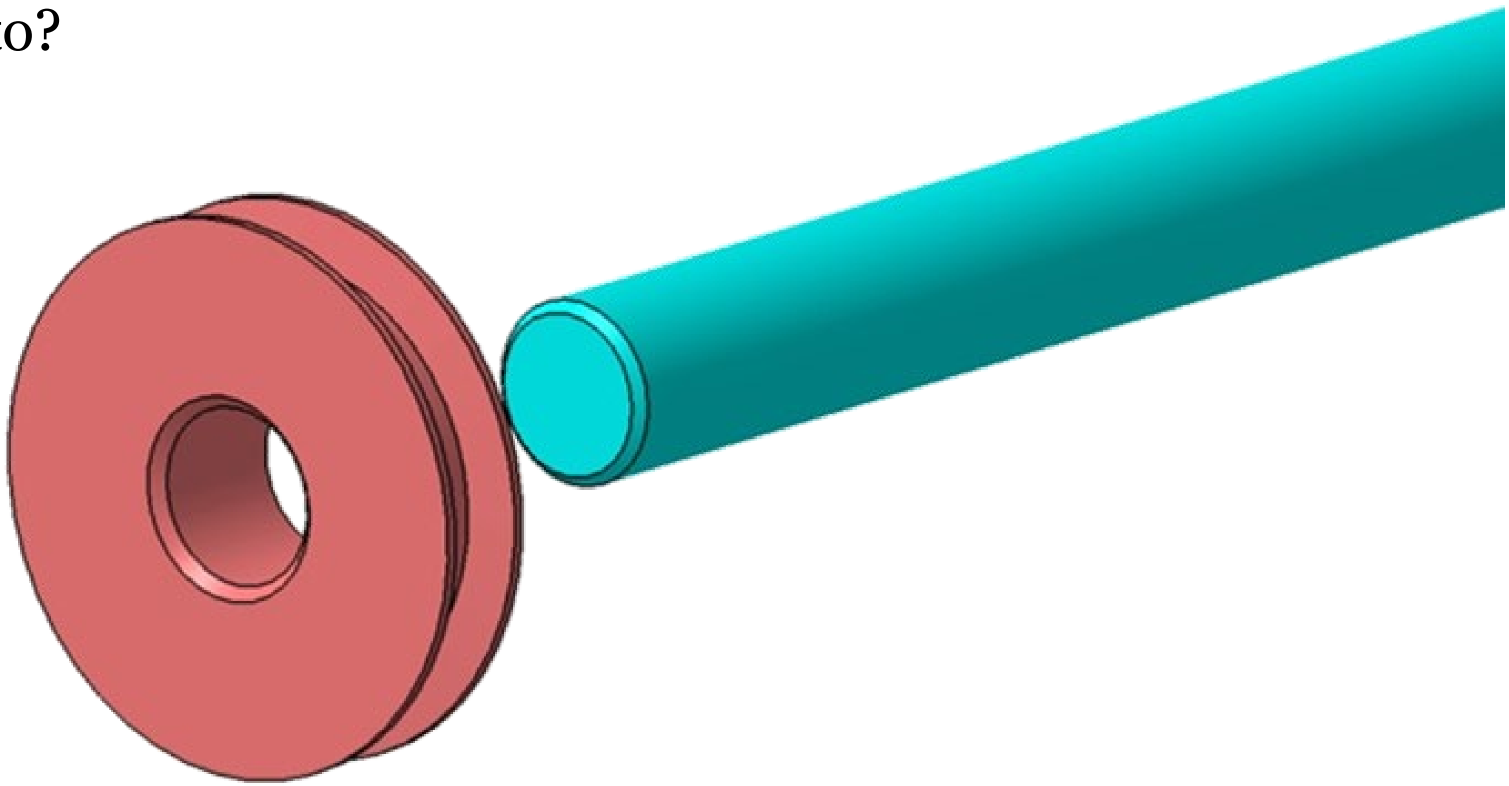
Transferring torque between components

Have you noticed that the shape of a shaft is round and that all pulleys and gears have holes in the centre for the shafts to fit into?



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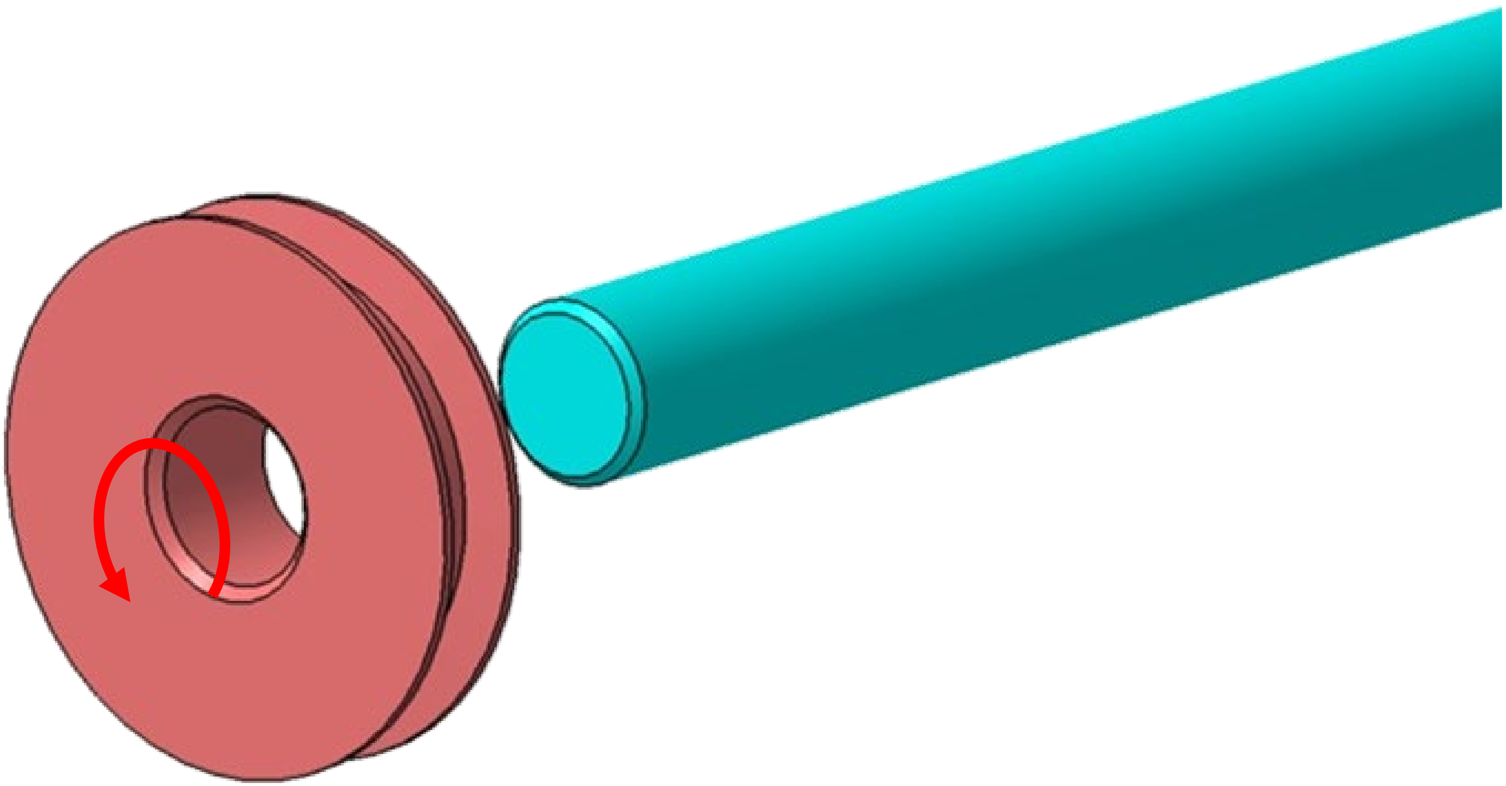
Transferring torque between components

How do we then prevent the torque that we wish to transfer from simply spinning the shaft within the pulleys or gears?



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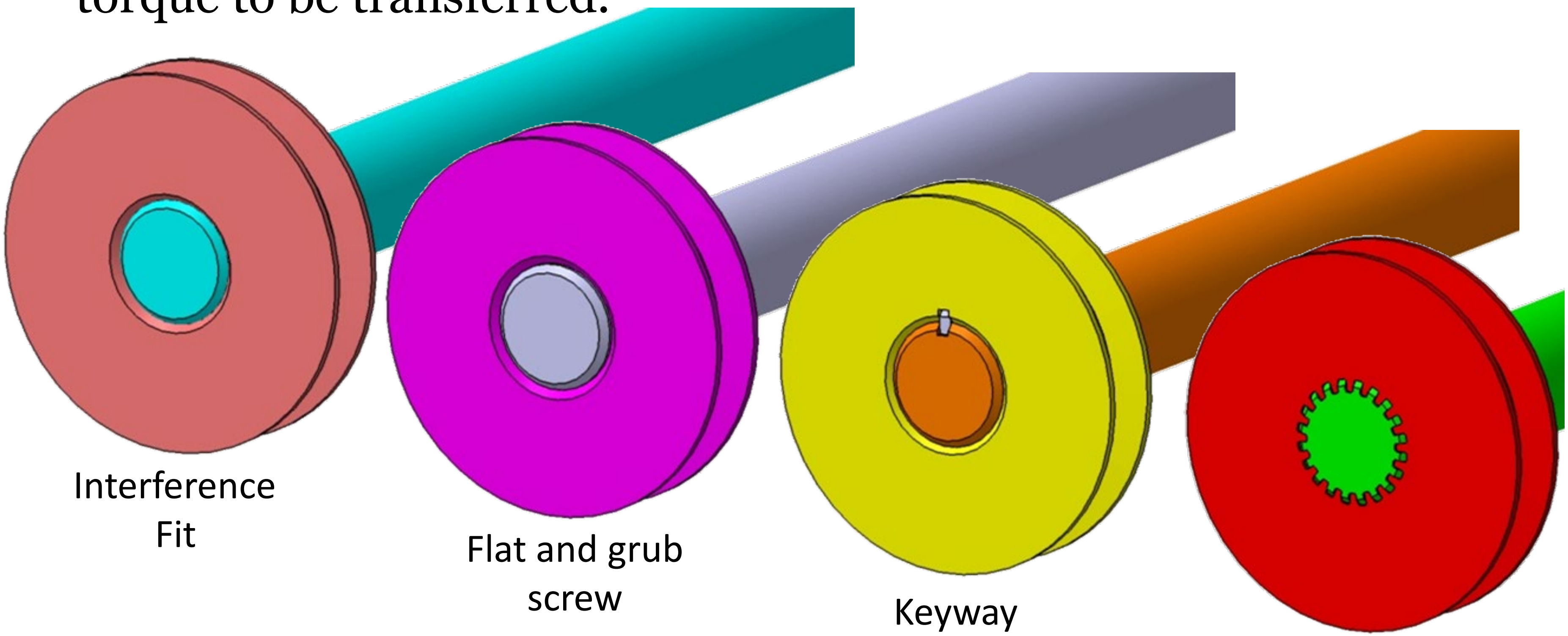
Transferring torque between components

We need to include additional features such as interference fits, flats, keyways and splines into our shaft design to allow the torque to be transferred.



Interference Fit

Flat and grub screw

Keyway

Splines

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DEPARTMENT OF ENGINEERING
DR SAMMY DIASINOS

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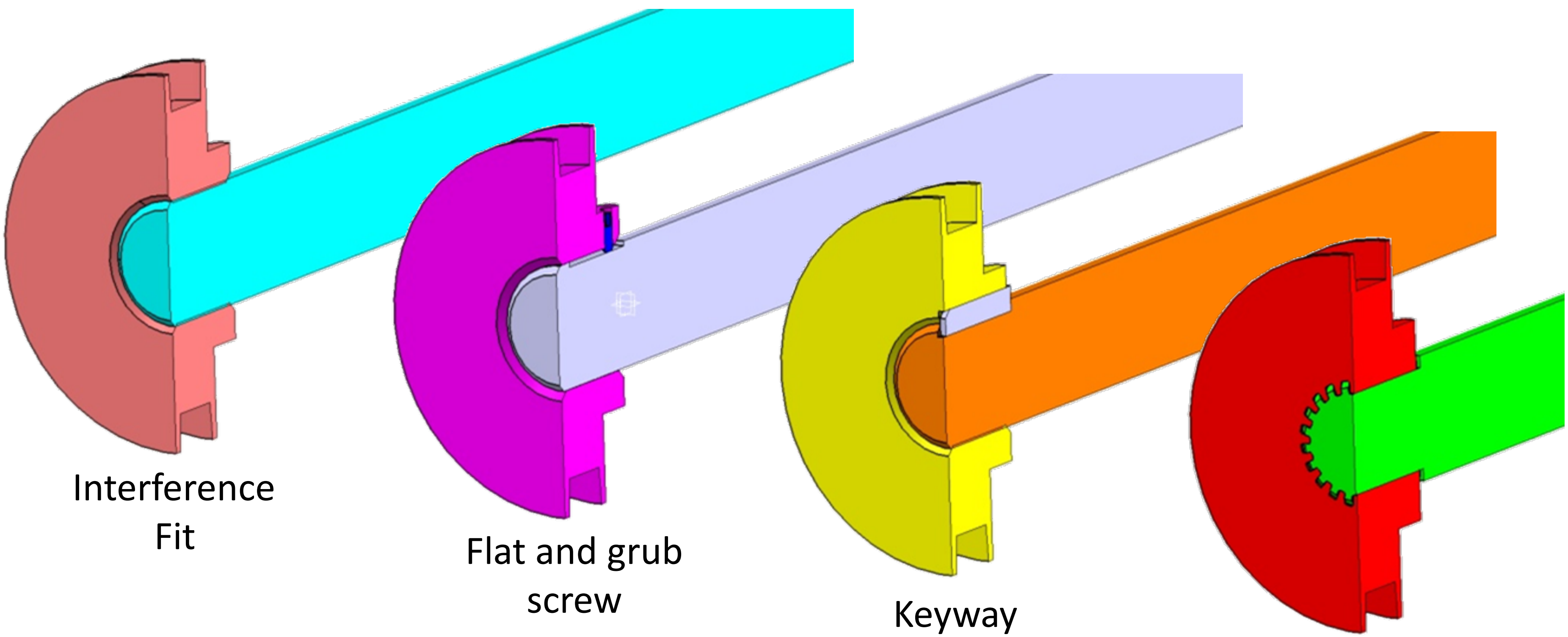
Transferring torque between components

In cross section we can see the geometry that allows for the torque to be transferred from the shaft to or from the pulley.



Interference Fit

Flat and grub screw

Keyway

Splines

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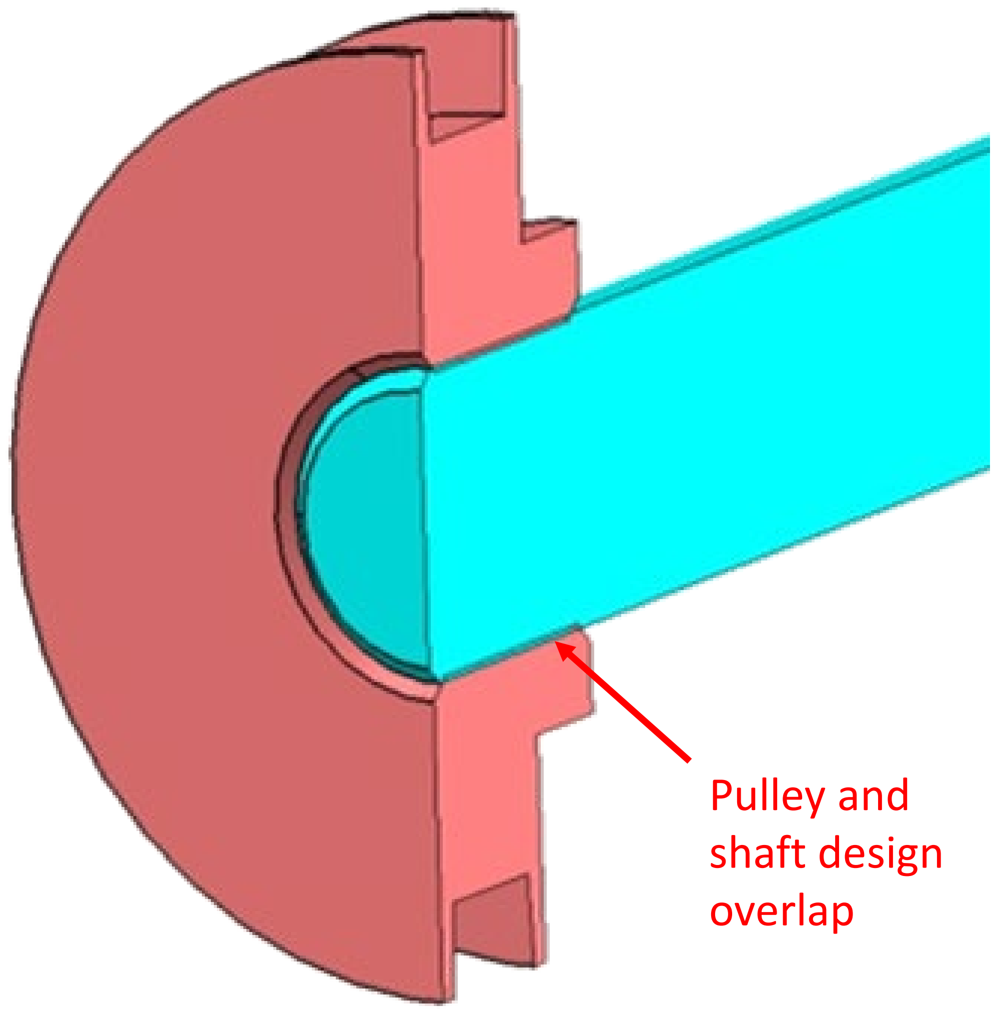
Transferring torque between components

The interference fit relies on a shaft which is oversized relative to the hole in the pulley/gear. To assemble this, it is necessary to heat the pulley and cool the shaft. Once all the components return to the same temperature they will apply pressure to each other.



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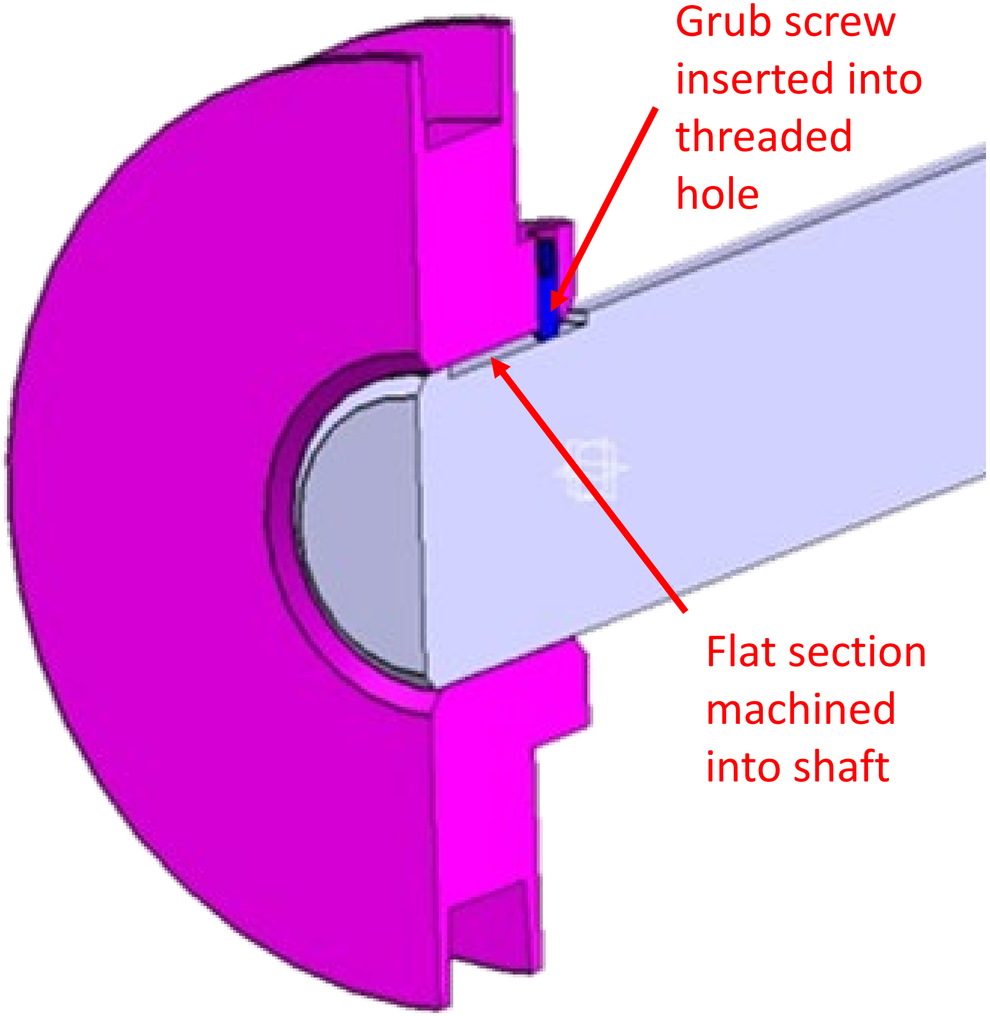
Transferring torque between components

By machining a flat portion into the shaft, it is possible to fix a pulley/gear to the shaft using a grub screw. To do this a threaded hole is required in the pulley/gear where the grub screw is inserted.



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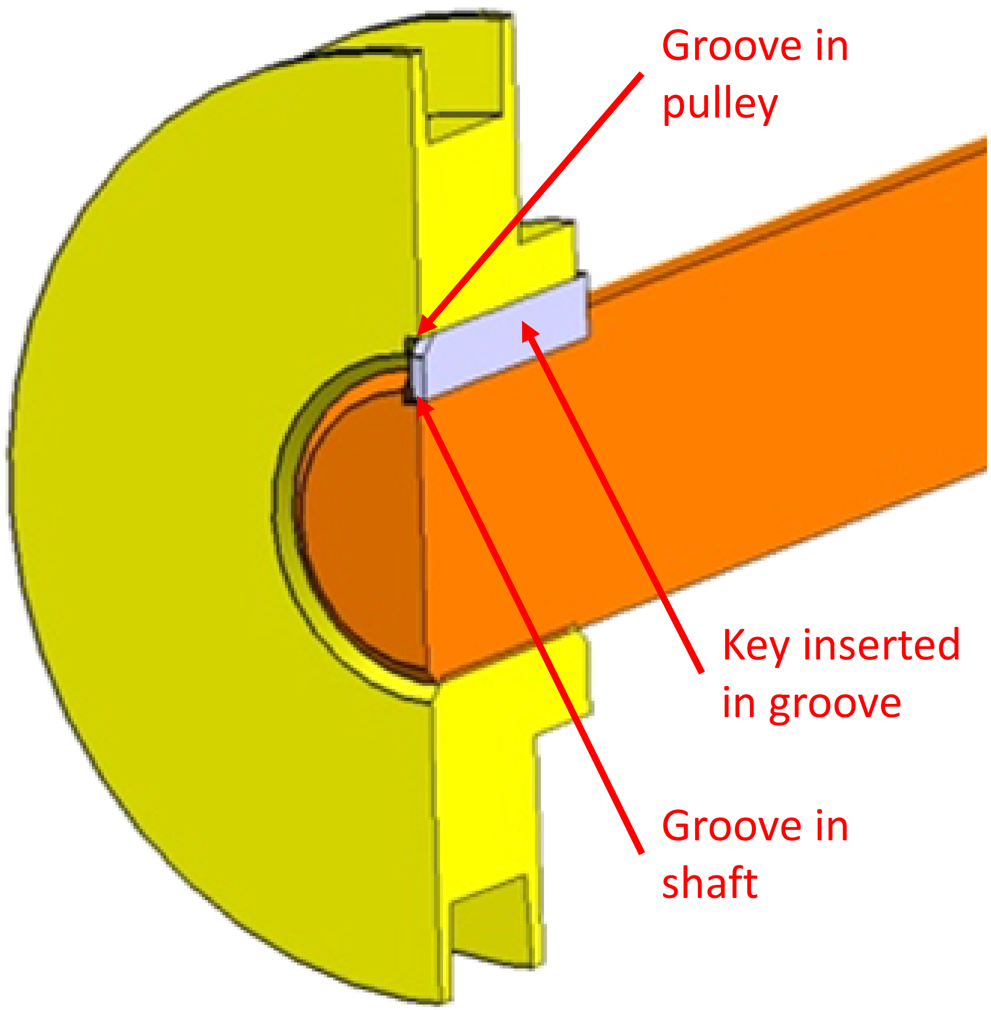
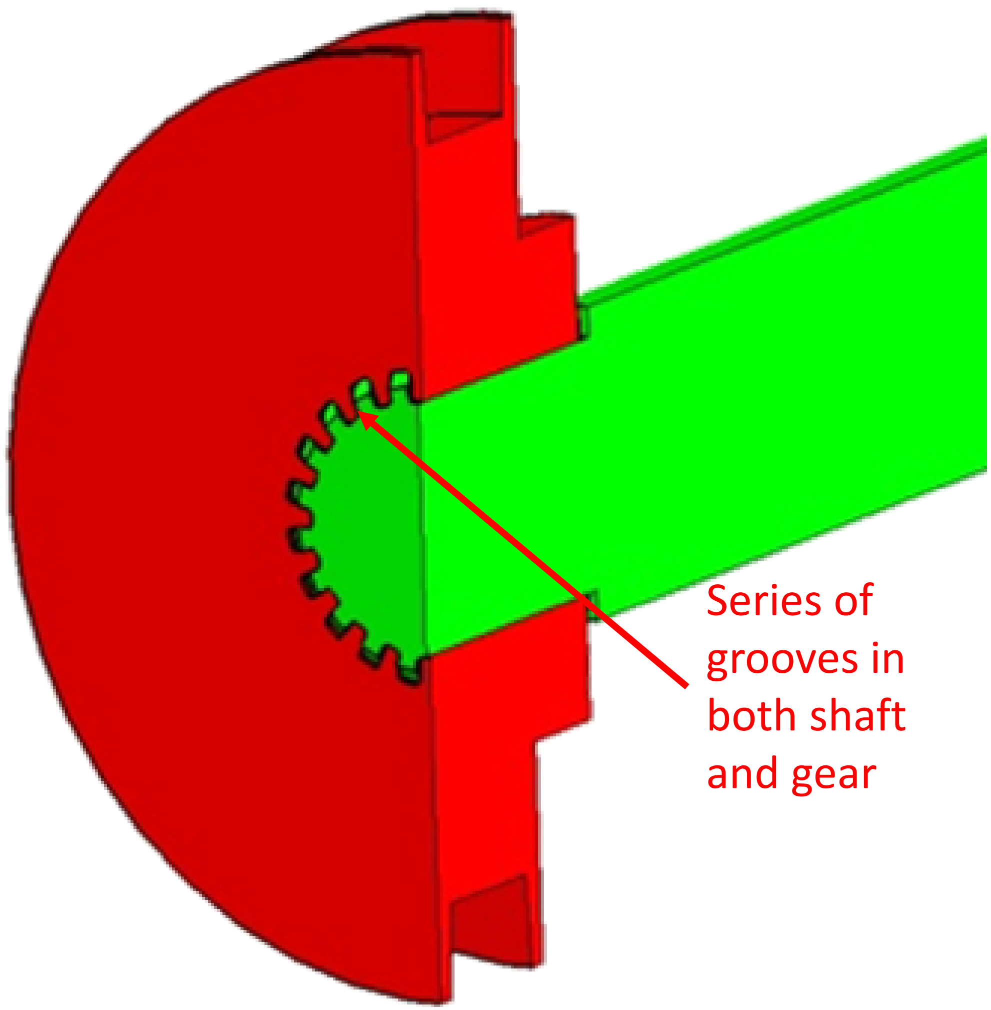
Transferring torque between components

A keyway requires a slot to be machined into the shaft and the pulley which a "key" can be inserted into preventing the two from rotating independently.



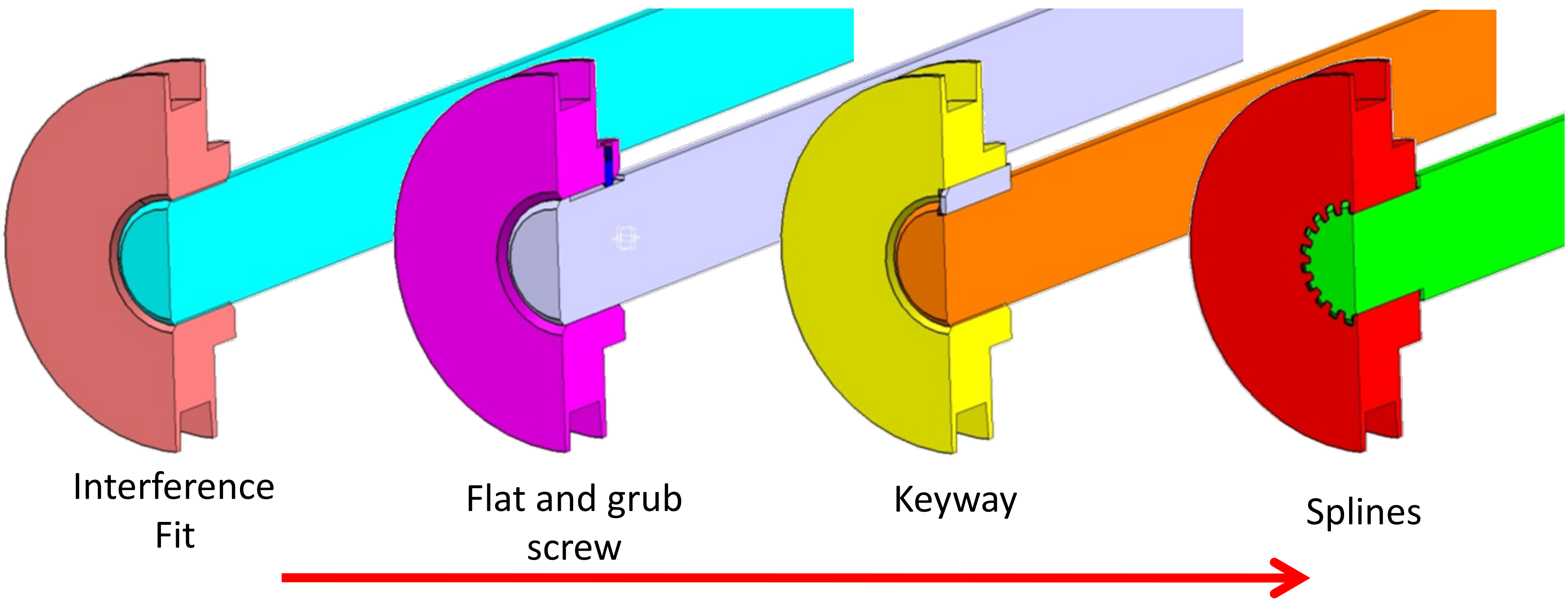
Transferring torque between components

Splines are a series of grooves or teeth that are machined into both the shaft and the pulley/gear, acting like many keyways.



Transferring torque between components

The option that you would use is a trade off between the cost of manufacture and the torque that is required to be transferred.



Least to most expensive and least torque transfer capability to most.

Example 9: I require to transfer large amounts of power between two shafts in a machine without changing the ratio of torque or angular velocity. The relative angular position of the two shafts is a high priority and they are located approximately 100cm apart. What mechanism should I use and how should they be connected to their respective shafts?

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Coupling and torque transfer selection

Example 10: I am mass producing a toy car that must be very cheap and require a method for transferring power from the electric motor to its drive axle. What components should I use and how should they be fixed to their respective shafts?

Coupling and torque transfer selection

Example 11: A design is required to transfer torque from the engine to the drive shafts that are at right angles for this drag car. What components should I use to transfer the power between the two shafts and how should these components be fixed to their respective shafts?

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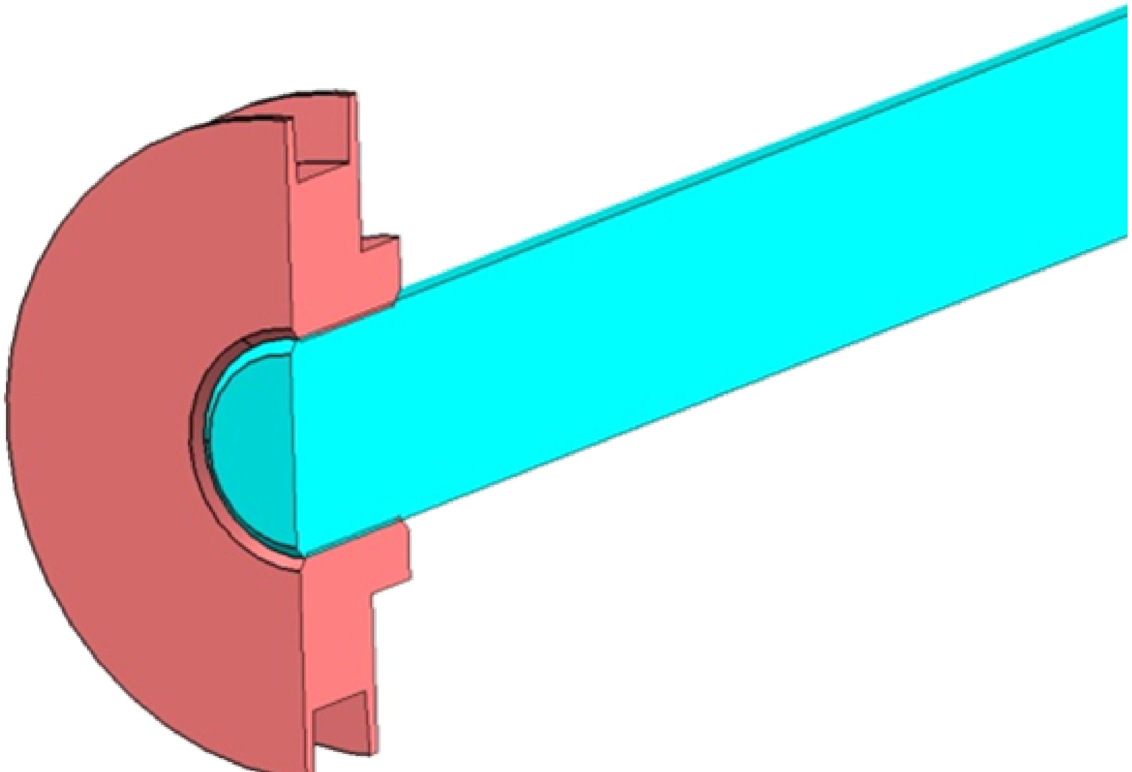
Tolerances

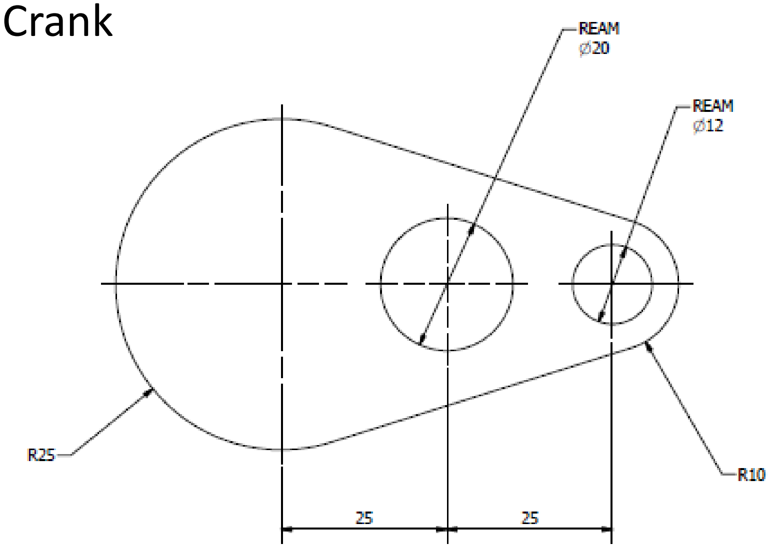
One of the ways that was identified to transfer torque between a shaft and a pulley/gear was through the use of an interference fit, but how much interference is required to achieve this?



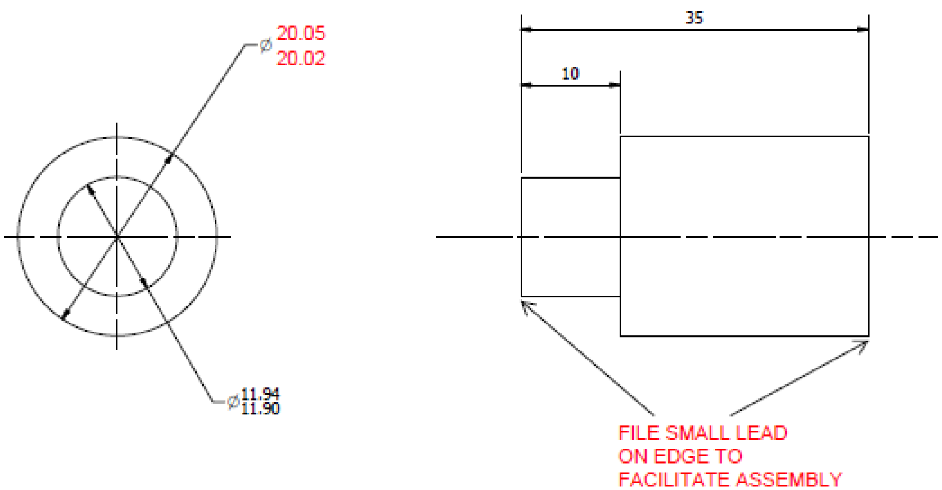
Tolerances

The level of interference required is actually quite small and is specified by the tolerance provided for that particular section of the shaft.

Crank



Crankshaft short



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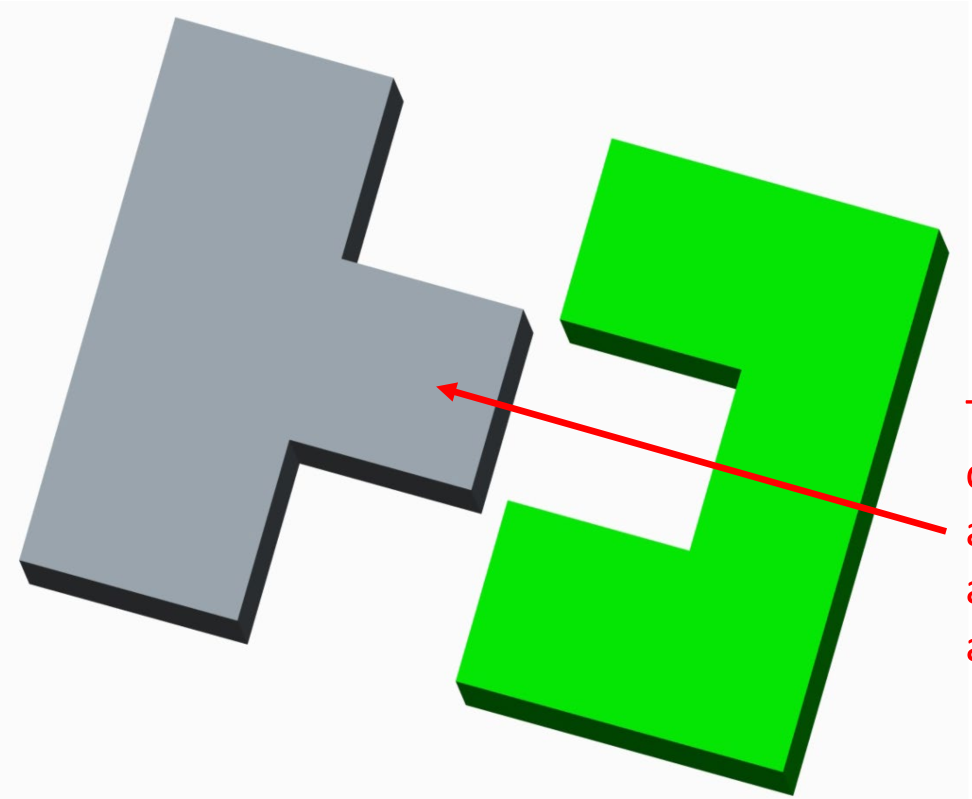
Tolerances

It should be noted that when discussing tolerances, the **shaft** refers to the component that fits into another member and does not only refer to circular sections.



This piece of the component is the shaft and its width requires a tolerance to achieve a desired fit.

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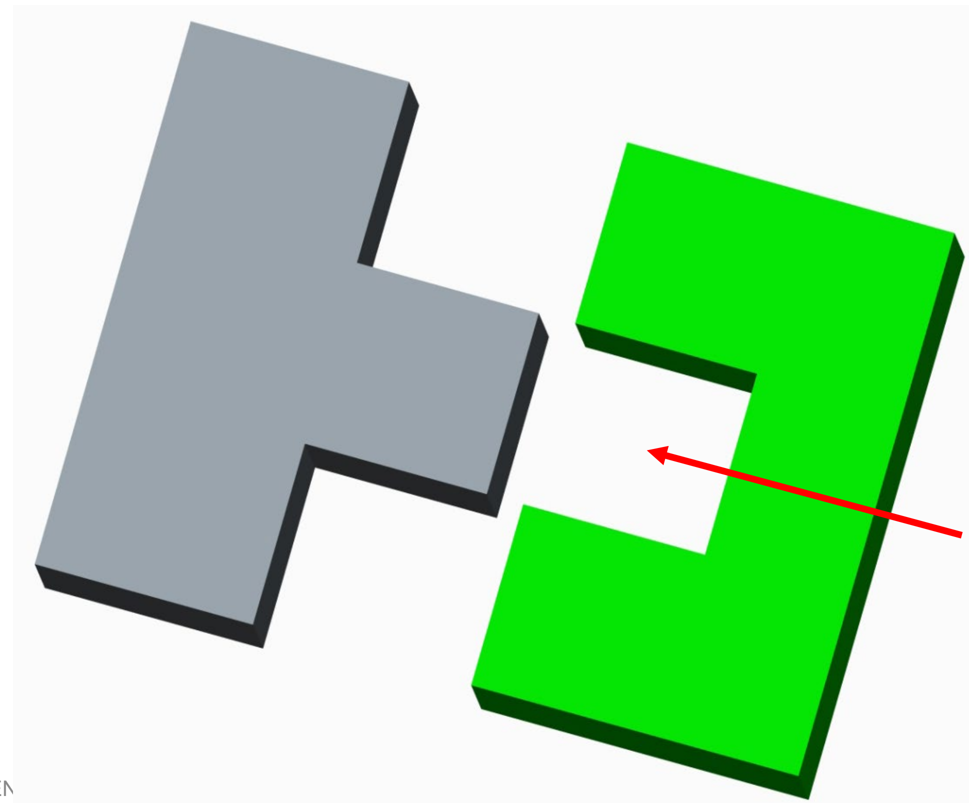
Tolerances

The **hole** is the location that the shaft fits into, this may not necessarily be literally a hole only, it can also be a gap between two other components or a slot.



This section of the component is the hole and its width requires a tolerance to achieve a desired fit.

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Their are alternative types of fits that can have tolerances assigned to them. The three main types are:

- Interference fits
- Transition fits
- Clearance fits

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- Interference fits *– The two components are fixed together due to the shaft being larger than the hole and can only be assembled after heating and cooling.*
- Transition fits
- Clearance fits

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Their are alternative types of fits that can have tolerances assigned to them. The three main types are:

- Interference fits *– The two components are fixed together due to the shaft being larger than the hole and can only be assembled after heating and cooling.*
- Transition fits *– The two components can be assembled with some effort and the two mating components are very similar sizes.*
- Clearance fits

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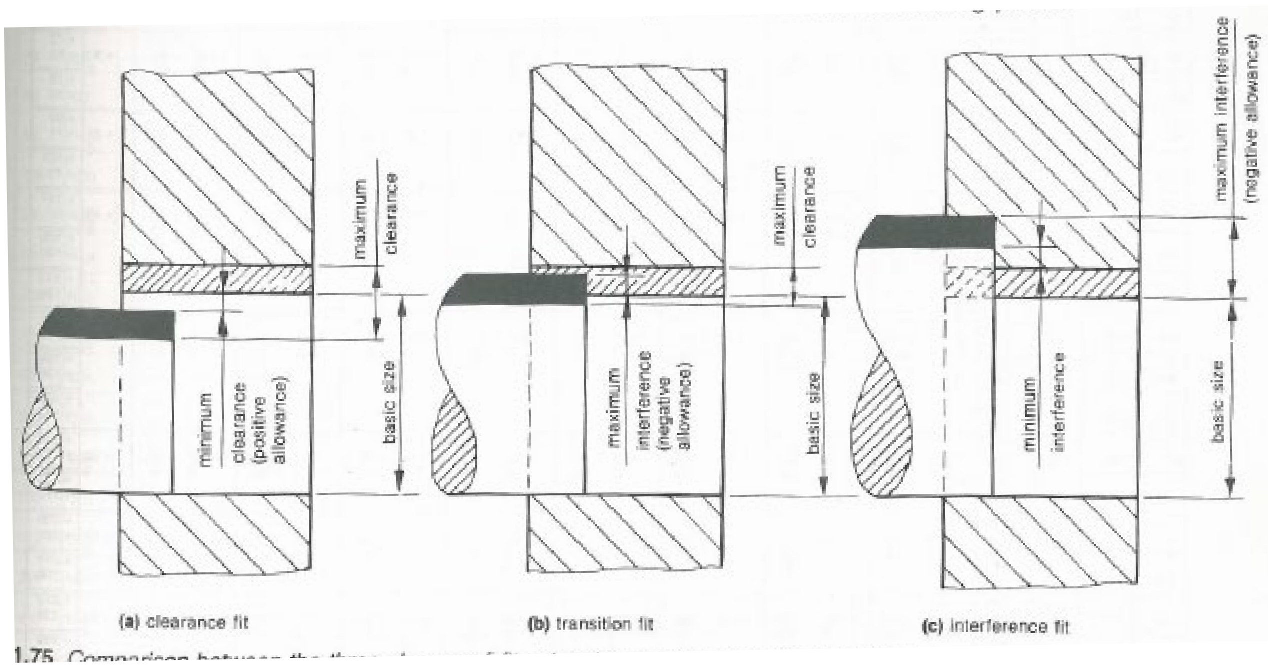
Their are alternative types of fits that can have tolerances assigned to them. The three main types are:

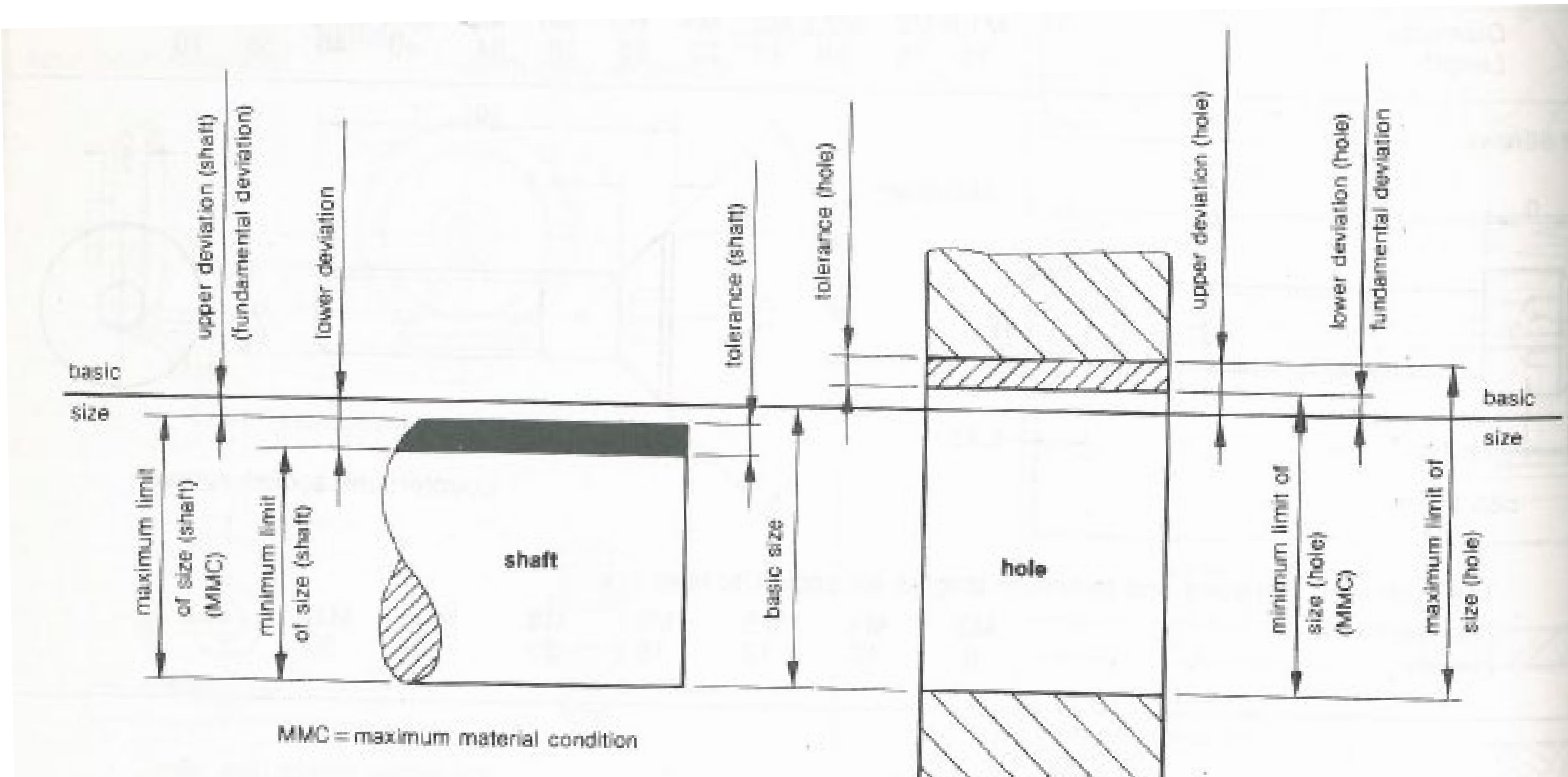
- Interference fits *– The two components are fixed together due to the shaft being larger than the hole and can only be assembled after heating and cooling.*
- Transition fits *– The two components can be assembled with some effort and the two mating components are very similar sizes.*
- Clearance fits *– The two components easily fit together as the hole is larger than the shaft.*

Pictorially, the three type of fits can be represented using the following diagram when assuming that the hole is maintained constant and the shaft size is altered to achieve the fit.



Tolerances

Tolerances for each fit include a **Basic size** as well as **Limits of size** which are depicted in the diagram below:



ig. 1.74 *Designation of shaft and hole sizes and limits*

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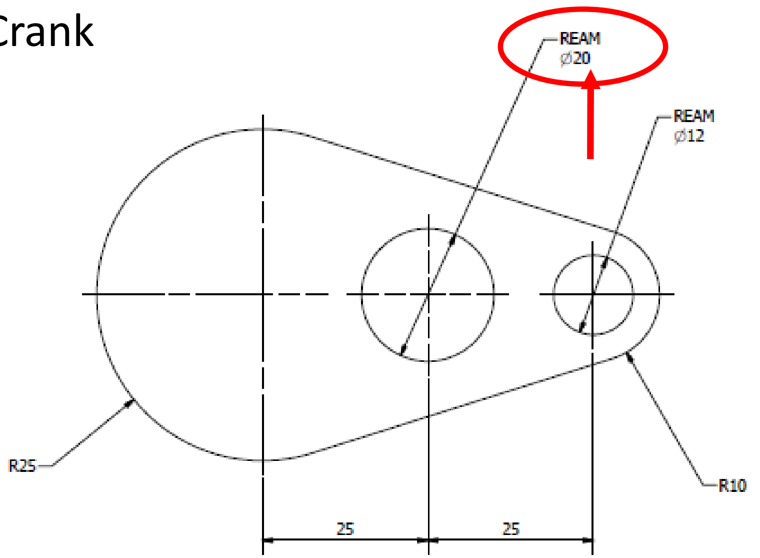
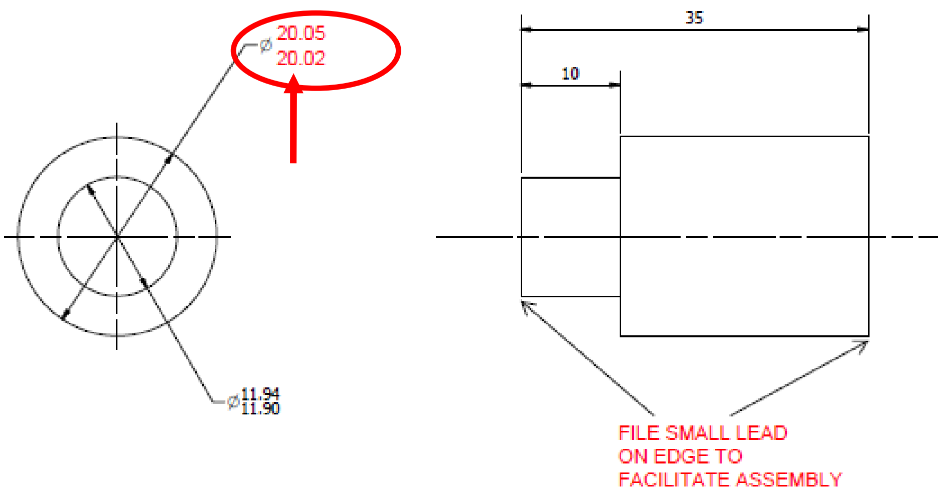
Tolerances

The **Basic size** is the value which the limits will be set around and is usually common between the hole and the shaft. For this example, the basic size is 20mm.

Crank



Crankshaft short



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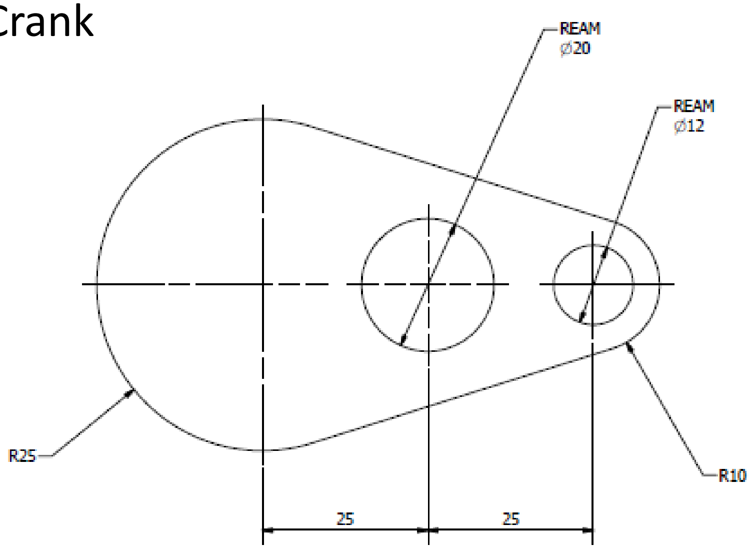
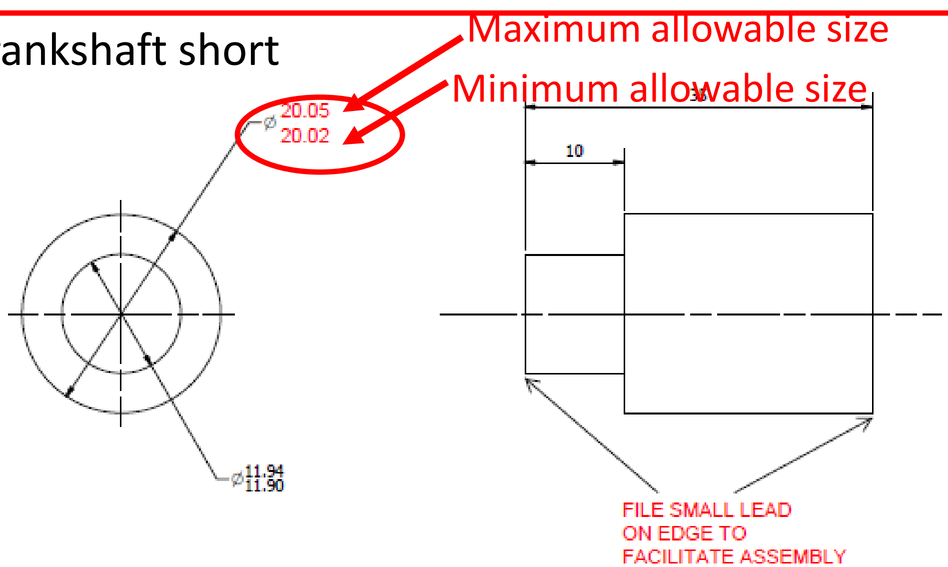
Tolerances

The **Limit of size** is the variation permitted from the basic size to achieve the desired tolerance. It includes both a maximum and minimum allowable size.

Crank



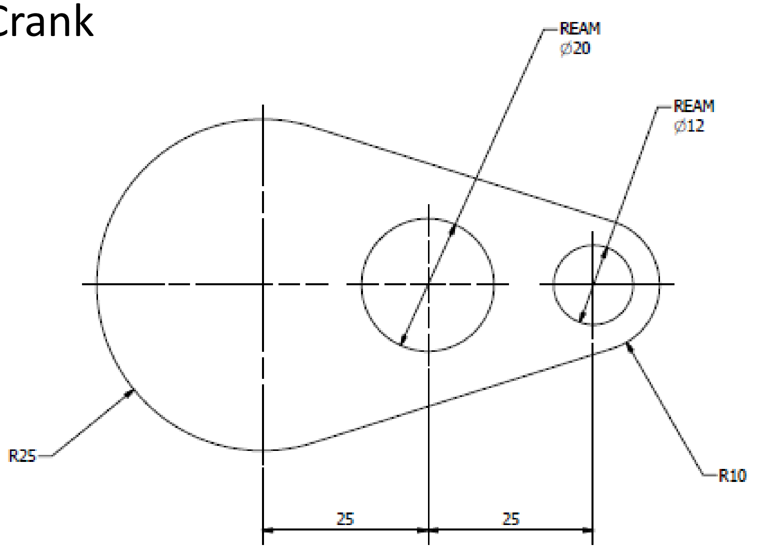
Crankshaft short



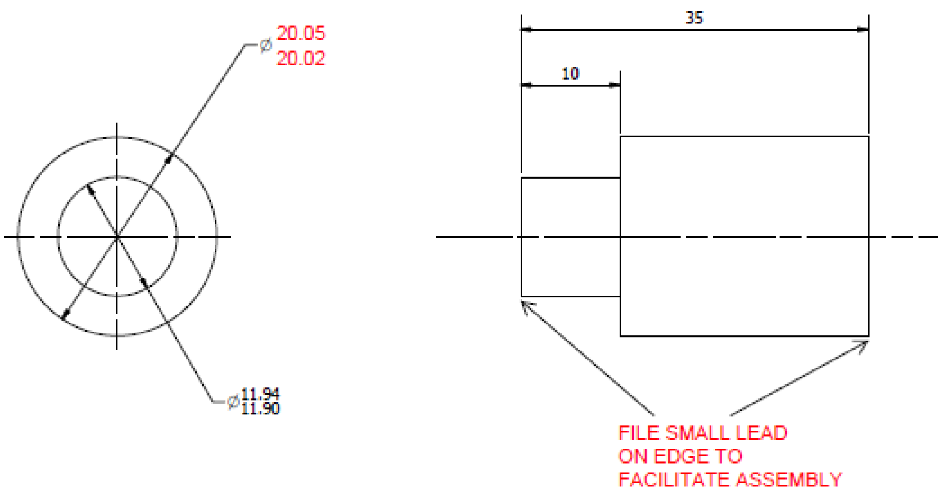
Tolerances

The **Upper** and **Lower Deviation** is the difference between the base size and the maximum or minimum limit size, in this case +0.05 and +0.02 respectively.

Crank



Crankshaft short



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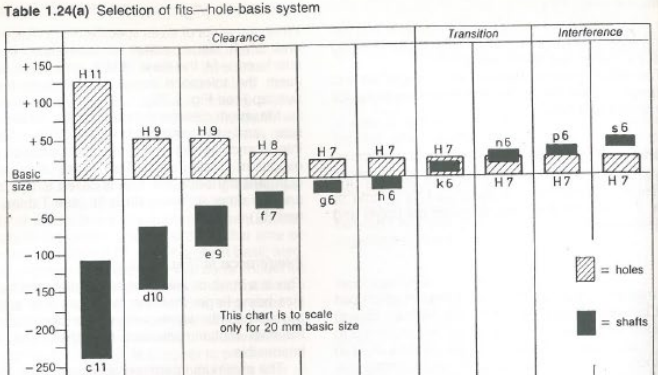
Tolerances

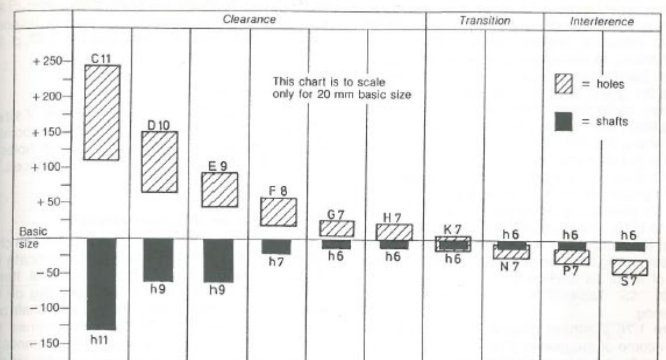
The values that should be used to achieve each of the individual fits are dependant on the basic size of the feature that is being designed and the desired fit. The tolerances can be found using tables such as these:

Table 1.24(a) Selection of fits—hole-basis system



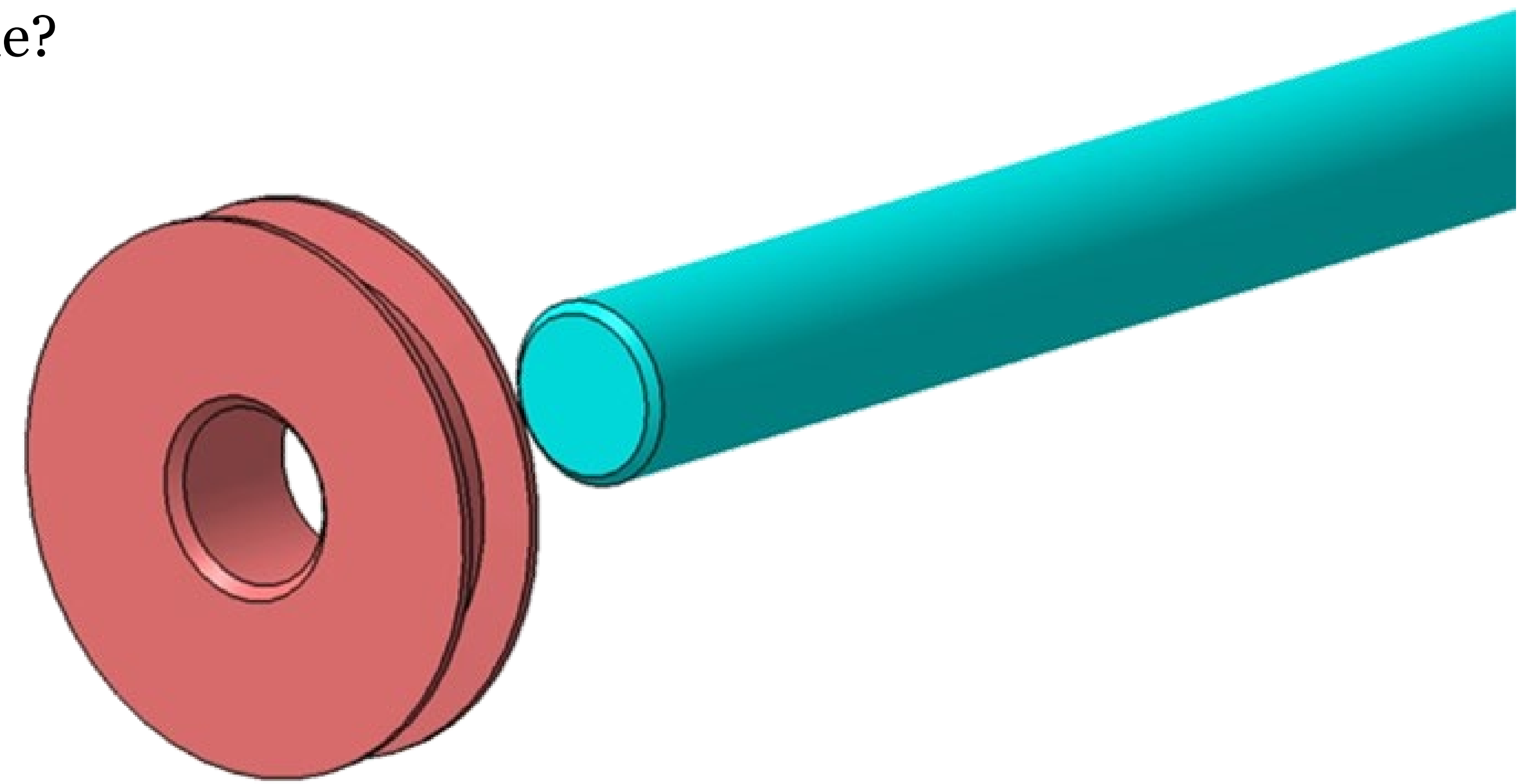
Tolerance unit = 0.001mm		Coarse tolerance	Loose (running fit)	Easy (running fit)	Normal running	Precision running, location	Average location	Light (push fit)	Heavy (push fit)	Press fit (ferrous)	Heavy press fit (non-ferrous)
Basic sizes (mm) over	to	H11–c11	H9–d10	H9–e9	H8–f7	H7–g6	H7–h6	H7–k6	H7–n6	H7–p6	H7–s6
0	3	+60 −60 0 −120	+25 −20 0 −60	+25 −14 0 −39	+14 −6 0 −16	+10 −2 0 −8	+10 0 0 −6	+10 +6 0 0	+10 +10 0 +4	+10 +12 0 +6	+10 +20 0 +14
3	6	+75 −70 0 −145	+30 −30 0 −78	+30 −20 0 −50	+18 −10 0 −28	+12 −4 0 −12	+12 0 0 −8	+12 +9 0 +1	+12 +16 0 +8	+12 +20 0 +12	+12 +27 0 +19
6	10	+90 −80 0 −170	+36 −40 0 −98	+36 −25 0 −61	+22 −13 0 −28	+15 −5 0 −14	+15 0 0 −9	+15 +10 0 +1	+15 +19 0 +10	+15 +24 0 +15	+15 +32 0 +23
10	18	+110 −95 0 −205	+43 −50 0 −120	+43 −32 0 −75	+27 −16 0 −34	+18 −6 0 −17	+18 0 0 −11	+18 +12 0 +1	+18 +23 0 +12	+18 +29 0 +18	+18 +39 0 +28
18	30	+130 −110 0 −240	+52 −65 0 −149	+52 −40 0 −92	+33 −20 0 −41	+21 −7 0 −20	+21 0 0 −13	+21 +15 0 +2	+21 +28 0 +15	+21 +35 0 +22	+21 +48 0 +35
30	40	+160 −120 −280	+62 −80 0 −180	+62 −50 0 −112	+39 −25 0 −50	+25 −9 0 −25	+25 0 0 −16	+25 +18 0 +2	+25 +33 0 +17	+25 +42 0 +26	+25 +59 0 +43
40	50	0 −130 −290									
50	65	+190 −140 −330	+74 −100 0 −220	+74 −60 0 −134	+46 −30 0 −60	+30 −10 0 −29	+30 0 0 −19	+30 +21 0 +2	+30 +39 0 +20	+30 +51 0 +32	+30 +72 +53
65	80	0 −150 −340									0 +78 +59
80	100	+220 −170 −390	+87 −120 0 −260	+87 −72 0 −159	+54 −36 0 −71	+35 −12 0 −34	+35 0 0 −22	+35 +25 0 +3	+35 +45 0 +23	+35 +59 0 +37	+35 +93 +71
		−180									0 +101 +79

Table 1.24(b) Selection of fits—shaft-basis system



Tolerance unit = 0.001mm		Coarse tolerance	Loose (running fit)	Easy (running fit)	Normal running	Precision running, location	Average location	Light (push fit)	Heavy (push fit)	Press fit (ferrous)	Heavy press fit (non-ferrous)
Basic sizes (mm) over	to	C11–h11	D10–h9	E9–h9	F8–h7	G7–h6	H7–h6	K7–h6	N7–h6	P7–h6	S7–h6
0	3	+120 0 +60 −60	+60 0 +20 −25	+39 0 +14 −25	+20 0 +6 −10	+12 0 +2 −6	+10 0 0 −6	0 0 −10 −6	−4 0 −14 −6	−6 0 −16 −6	−14 0 −24 −6
3	6	+145 0 +70 −75	+78 0 +30 −30	+50 0 +20 −30	+28 0 +10 −12	+16 0 +4 −8	+12 0 0 −8	+3 0 −9 −8	−4 0 −16 −8	−8 0 −20 −8	−15 0 −27 −8
6	10	+170 0 +80 −90	+98 0 +40 −36	+61 0 +25 −36	+35 0 +13 −15	+20 0 +5 −9	+15 0 0 −9	+5 0 −10 −9	−4 0 −19 −9	−9 0 −24 −9	−17 0 −32 −9
10	18	+205 0 +95 −110	+120 0 +50 −43	+75 0 +32 −43	+43 0 +16 −18	+24 0 +6 −11	+18 0 0 −11	+6 0 −12 −11	−5 0 −23 −11	−11 0 −29 −11	−21 0 −39 −11
18	30	+240 0 +110 −130	+149 0 +65 −52	+92 0 +40 −52	+53 0 +20 −21	+28 0 +7 −13	+21 0 0 −13	+6 0 −15 −13	−7 0 −28 −13	−14 0 −35 −13	−27 0 −48 −13
30	40	+280 0 +120 −160	+180 0 +80 −62	+112 0 +50 −62	+64 0 +25 −25	+34 0 +9 −16	+25 0 0 −16	+7 0 −18 −16	−8 0 −33 −16	−17 0 −42 −16	−34 0 −59 −16
40	50	+290 0 +130 −160									
50	65	+330 0 +140 −190	+220 0 +100 −74	+134 0 +60 −74	+76 0 +30 −30	+40 0 +10 −19	+30 0 0 −19	+9 0 −21 −19	−9 0 −39 −19	−21 0 −51 −19	−42 0 −72 −19
65	80	+340 0 +150 −190									−48 0 −78 −19
80	100	+390 0 +170 −220	+260 0 +120 −87	+159 0 +72 −87	+90 0 +36 −35	+47 0 +12 −22	+35 0 0 −22	+10 0 −25 −22	−10 0 −45 −22	−24 0 −59 −22	−58 0 −93 −22
100	120	+400 0 [illegible]									−66 0

Example 12: The following shaft and pulley is made of steel, if the shaft and hole diameter are 45mm, and I wish to have an interference fit, what tolerances should I specify for the shaft and hole?



Tolerances

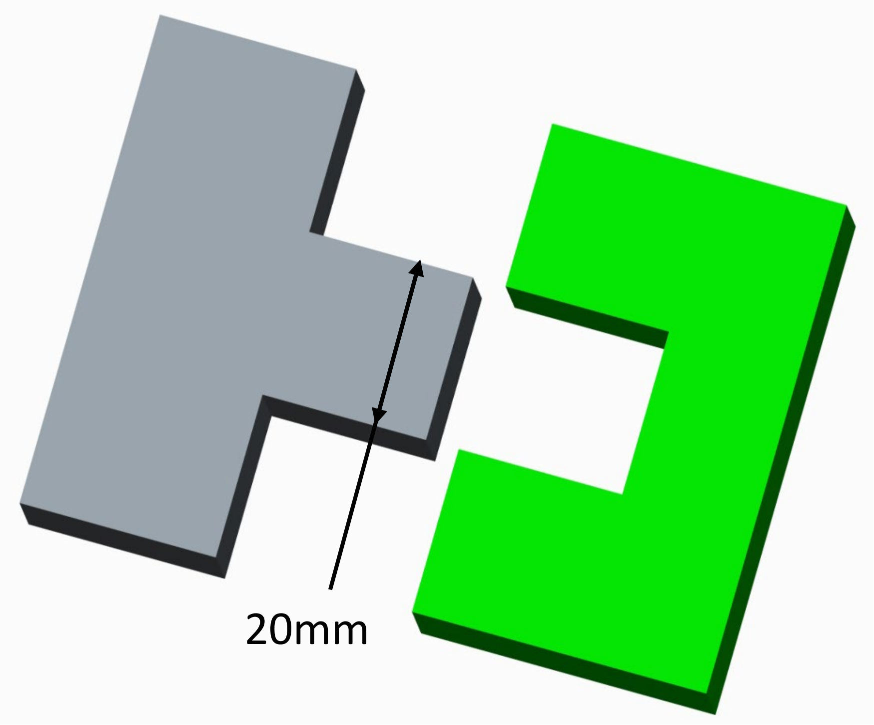
Example 12: The following shaft and pulley is made of steel, if the shaft and hole diameter are 45mm, and I wish to have an interference fit, what tolerances should I specify for the shaft and hole?

Tolerance (unit = 0.001mm)		Coarse tolerance		Loose running fit		Easy running fit		Normal running		Precision running, location		Average location		Light push fit		Heavy push fit		Press fit (ferrous)		Heavy press fit (non-ferrous)	
Basic sizes (mm)		H11–c11		H9–d10		H9–e9		H8–f7		H7–g6		H7–h6		H7–k6		H7–n6		H7–p6		H7–s6	
over	to																				
0	3	+60 0	−60 −120	+25 0	−20 −60	+25 0	−14 −39	+14 0	−6 −16	+10 0	−2 −8	+10 0	0 −6	+10 0	+6 0	+10 0	+10 +4	+10 0	+12 +6	+10 0	+20 +14
3	6	+75 0	−70 −145	+30 0	−30 −78	+30 0	−20 −50	+18 0	−10 −28	+12 0	−4 −12	+12 0	0 −8	+12 0	+9 +1	+12 0	+16 +8	+12 0	+20 +12	+12 0	+27 +19
6	10	+90 0	−80 −170	+36 0	−40 −98	+36 0	−25 −61	+22 0	−13 −28	+15 0	−5 −14	+15 0	0 −9	+15 0	+10 +1	+15 0	+19 +10	+15 0	+24 +15	+15 0	+32 +23
10	18	+110 0	−95 −205	+43 0	−50 −120	+43 0	−32 −75	+27 0	−16 −34	+18 0	−6 −17	+18 0	0 −11	+18 0	+12 +1	+18 0	+23 +12	+18 0	+29 +18	+18 0	+39 +28
18	30	+130 0	−110 −240	+52 0	−65 −149	+52 0	−40 −92	+33 0	−20 −41	+21 0	−7 −20	+21 0	0 −13	+21 0	+15 +2	+21 0	+28 +15	+21 0	+35 +22	+21 0	+48 +35
30	40	+160 0	−120 −280	+62 0	−80 −180	+62 0	−50 −112	+39 0	−25 −50	+25 0	−9 −25	+25 0	0 −16	+25 0	+18 +2	+25 0	+33 +17	+25 0	+42 +26	+25 0	+59 +43
40	50		−130 −290																		
50	65	+190 0	−140 −330	+74 0	−100 −220	+74 0	−60 −134	+46 0	−30 −60	+30 0	−10 −29	+30 0	0 −19	+30 0	+21 +2	+30 0	+39 +20	+30 0	+51 +32	+30 0	+72 +53
65	80		−150 −340																		+78 +59

MECH2003: Mechanical Design

Tolerances

Example 13: The following two components are required to be assembled and dissembled easily while being located accurately relative to each other. What tolerances should be applied to the hole and shaft?



MECH2003: Mechanical Design

Tolerances

Example 13: The following two components are required to be assembled and dissembled easily while being located accurately relative to each other. What tolerances should be applied to the hole and shaft?

Tolerance unit = 0.001mm		Coarse clearance	Loose running fit	Easy running fit	Normal running	Precision running, location	Average location	Light push fit	Heavy push fit	Press fit (ferrous)	Heavy press fit (non-ferrous)
Basic sizes (mm) over	to	$H11$–$c11$	$H9$–$d10$	$H9$–$e9$	$H8$–$f7$	$H7$–$g6$	$H7$–$h6$	$H7$–$k6$	$H7$–$n6$	$H7$–$p6$	$H7$–$s6$
0	3	+60 −60 0 −120	+25 −20 0 −60	+25 −14 0 −39	+14 −6 0 −16	+10 −2 0 −8	+10 0 0 −6	+10 +6 0 0	+10 +10 0 +4	+10 +12 0 +6	+10 +20 0 +14
3	6	+75 −70 0 −145	+30 −30 0 −78	+30 −20 0 −50	+18 −10 0 −28	+12 −4 0 −12	+12 0 0 −8	+12 +9 0 +1	+12 +16 0 +8	+12 +20 0 +12	+12 +27 0 +19
6	10	+90 −80 0 −170	+36 −40 0 −98	+36 −25 0 −61	+22 −13 0 −28	+15 −5 0 −14	+15 0 0 −9	+15 +10 0 +1	+15 +19 0 +10	+15 +24 0 +15	+15 +32 0 +23
10	18	+110 −95 0 −205	+43 −50 0 −120	+43 −32 0 −75	+27 −16 0 −34	+18 −6 0 −17	+18 0 0 −11	+18 +12 0 +1	+18 +23 0 +12	+18 +29 0 +18	+18 +39 0 +28
18	30	+130 −110 0 −240	+52 −65 0 −149	+52 −40 0 −92	+33 −20 0 −41	+21 −7 0 −20	+21 0 0 −13	+21 +15 0 +2	+21 +28 0 +15	+21 +35 0 +22	+21 +48 0 +35
30	40	+160 −120 0 −280	+62 −80 0 −180	+62 −50 0 −112	+39 −25 0 −50	+25 −9 0 −25	+25 0 0 −16	+25 +18 0 +2	+25 +33 0 +17	+25 +42 0 +26	+25 +59 0 +43
40	50	−130 −290									
50	65	+190 −140 0 −330	+74 −100 0 −220	+74 −60 0 −134	+46 −30 0 −60	+30 −10 0 −29	+30 0 0 −19	+30 +21 0 +2	+30 +39 0 +20	+30 +51 0 +32	+30 +72 0 +53
65	80	−150 −340									+78 +59

FACULTY OF SCIENCE AND ENGINEERING
DEPARTMENT OF ENGINEERING

DR SAMMY DIASINOS