

Transmissions and couplings:

Direct transmissions, levers, gears, toothed belts, screw-nut, kinematic transmissions

Composition of a machine (robot) axis.....

Motor



Couplings



Transmission Stages



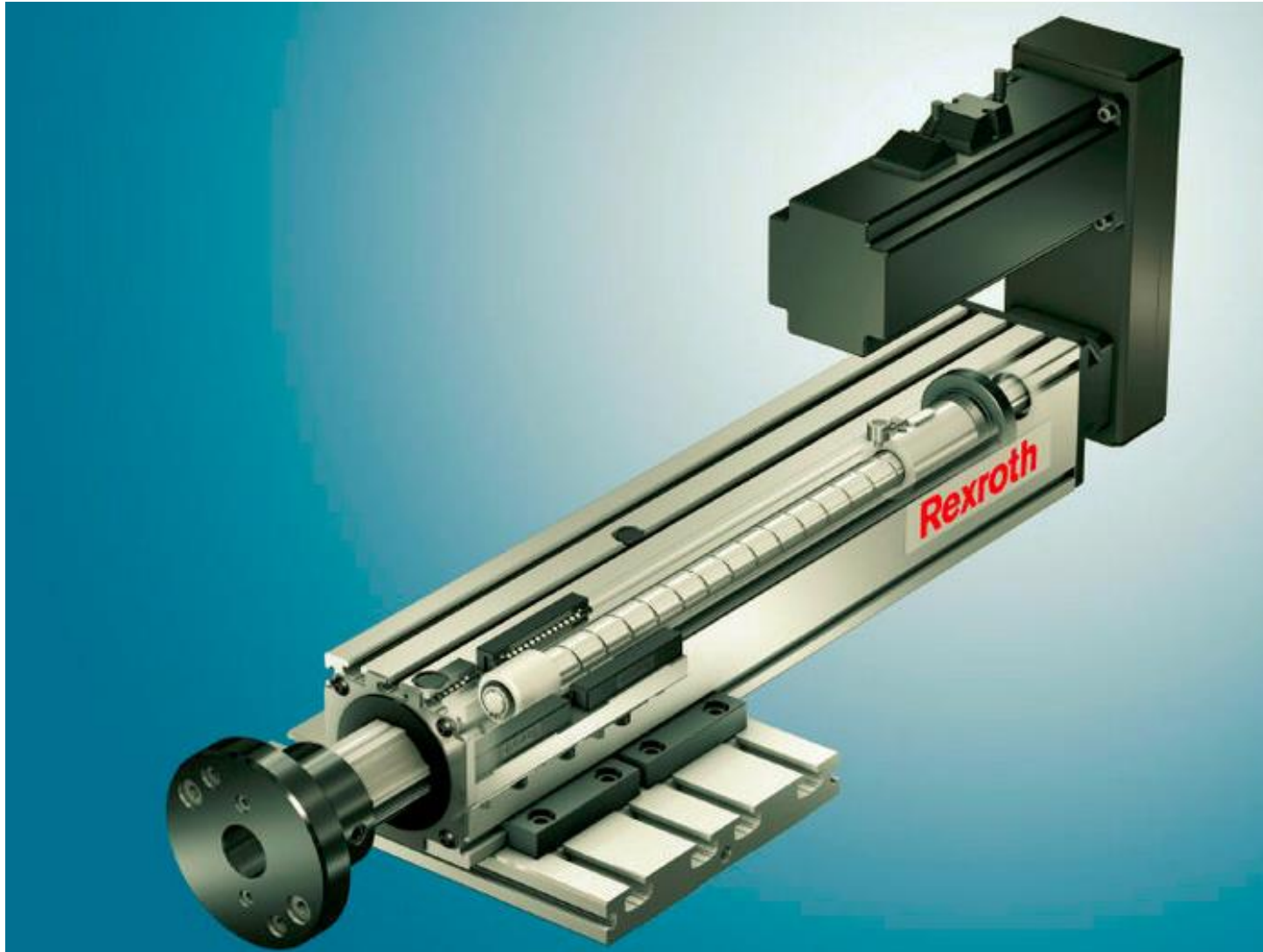
Guideways



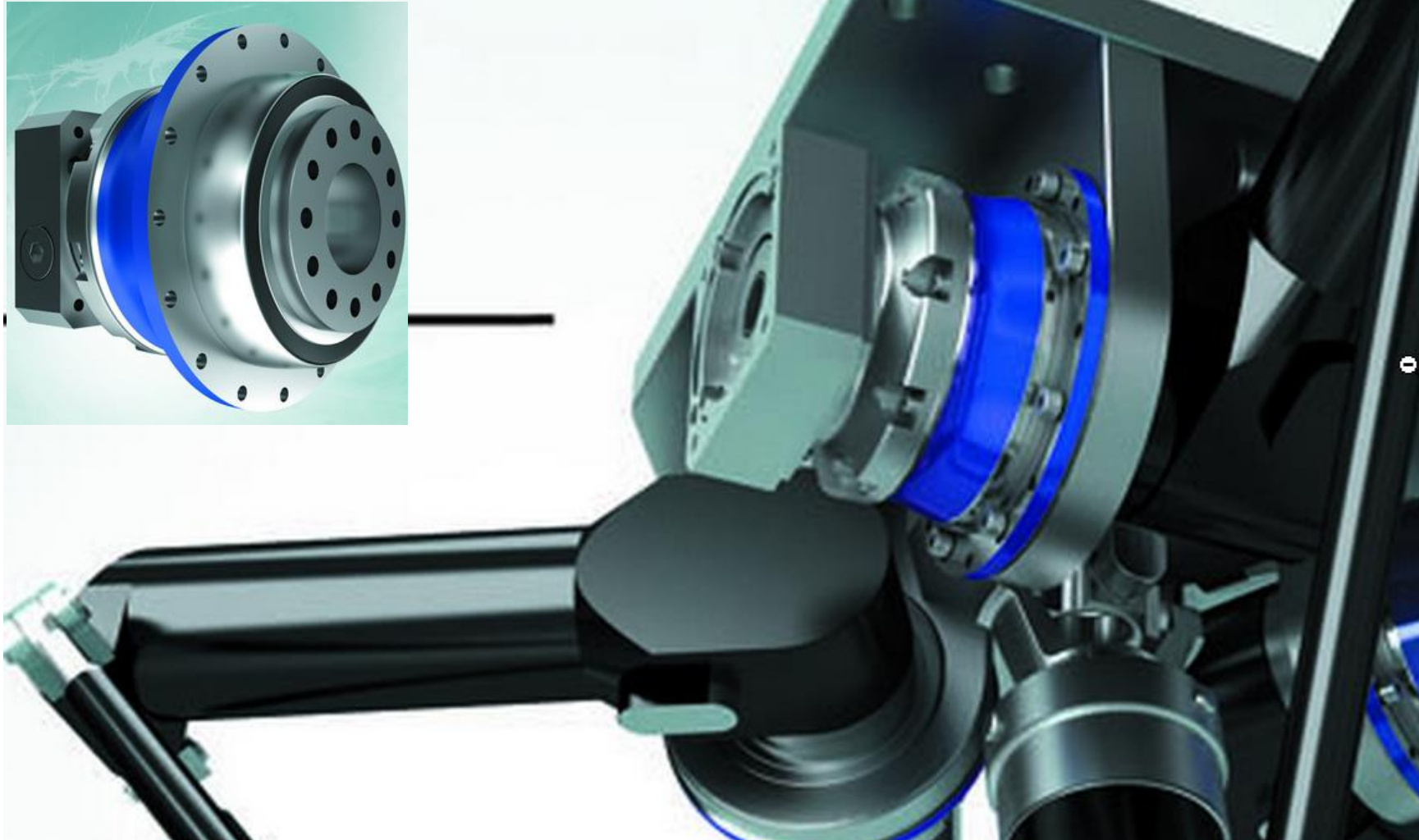
Desired characteristics for a transmission in a robot:

- High **stiffness** $\left(\frac{dM}{d\theta} \text{ or } \frac{dF}{dx} \text{ large} \right)$
- No **play** (ideally of course)
- Low **moment of inertia**
- Low **mass**

Example 1



Example 2: Rotational Arm



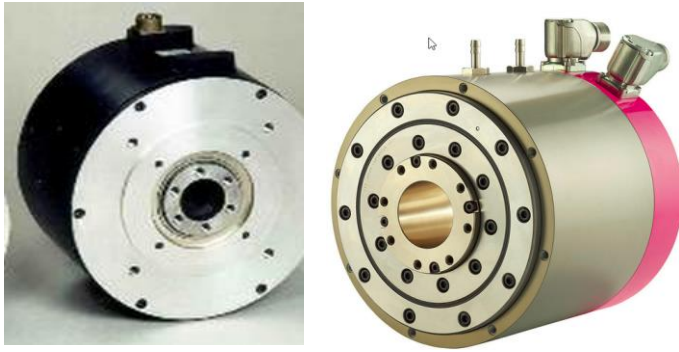
1- Direct Transmissions

In the case of a direct transmission, there is **no gearbox device** between the actuator (the motor) and the output.

A coupling is often used to combine (couple) the motion of the motor with the motion of the output shaft.

We will discuss **direct drive actuation**

- It is in any case a direct drive if the Motor is linear.



Hollow shaft torque motors.
The output arm is either screwed onto the rotating part of the motor or via a conical coupling



Torque motors with external shafts. The output arm is either clamped on the motor shaft or fixed by a conical coupling.

2- Couplings

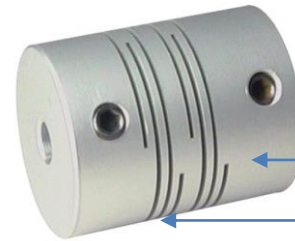
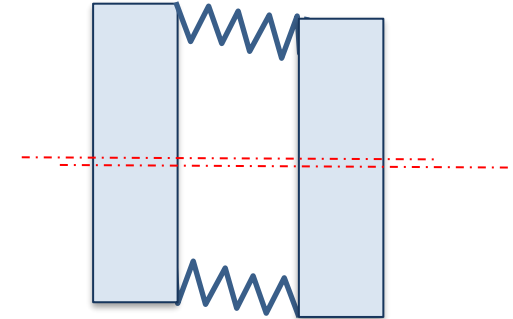
2.2 Direct fixation



2.2 The bellows (flexible coupling)

Allow to couple two shafts together and to correct the misalignment between the two rotation axes.

Will go up to 3-5Nm



Separation of bending weaknesses to correct orientations

The bellows with flexible blades (flexible coupling)

Will go up to 70-100 Nm

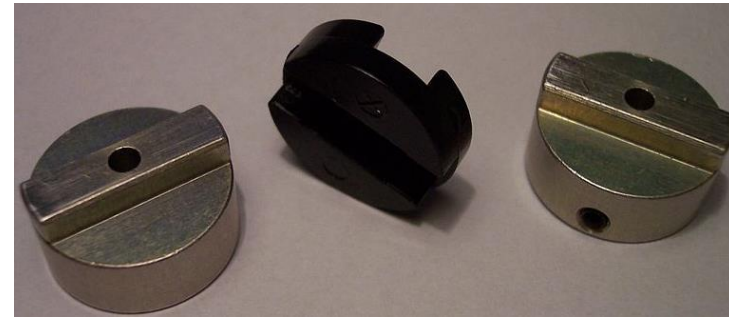


2.3 The Oldham system

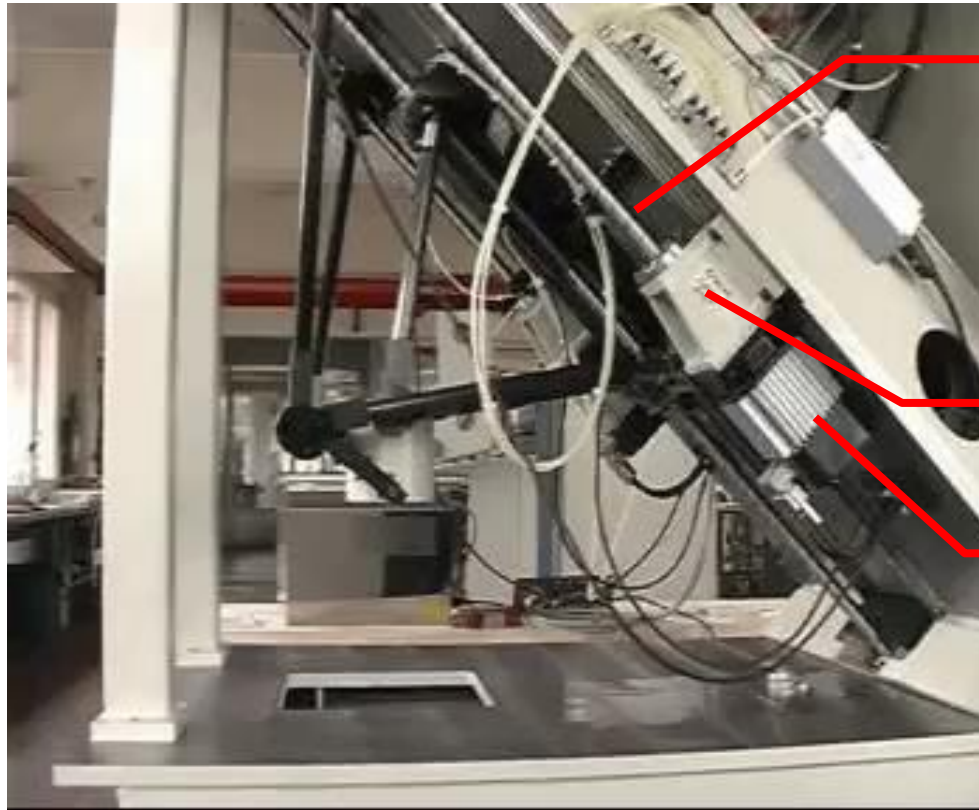
Named after John Oldham [Ireland 1821].
This coupling is simple but also inexpensive.

To be absolutely avoided in applications requiring precision.

Tolerated for speed control applications.



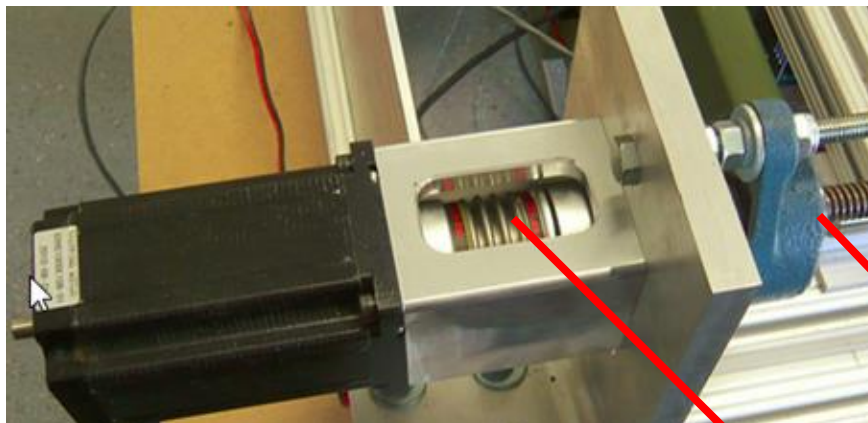
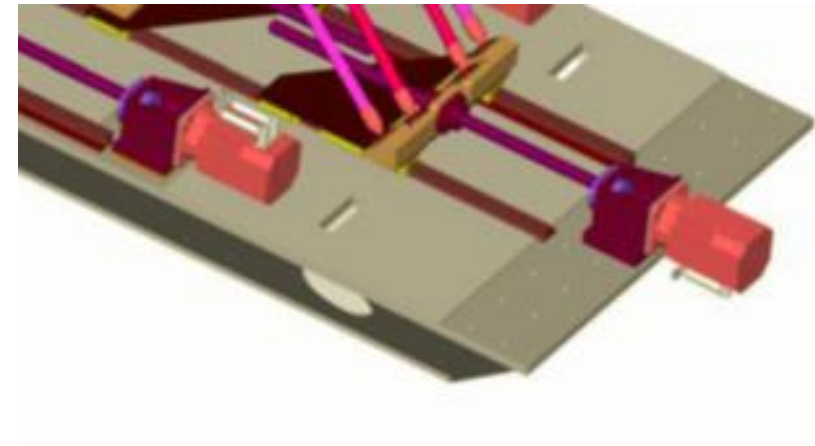
Example



Screw with 25mm pitch

Direct motor-screw
coupling by bellows

Motor 4 Nm

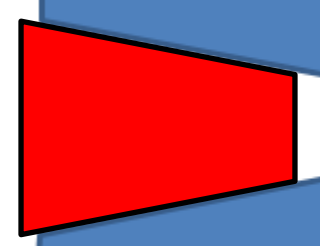


Double bearing

Coupling Bellow

2.3 The conical coupling

For the coupling of a hollow arm (or gear element) with a motor shaft, the conical coupling is an almost obvious solution when high torques (**tens of NM to hundreds of NM**) are to be transmitted.



The principle is that by pressing the cone against the reference part, the material deforms and the contact is ensured by friction.

Compression coupling of materials



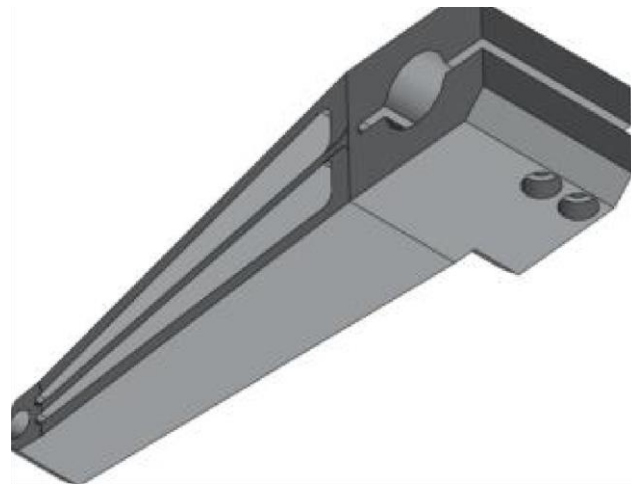
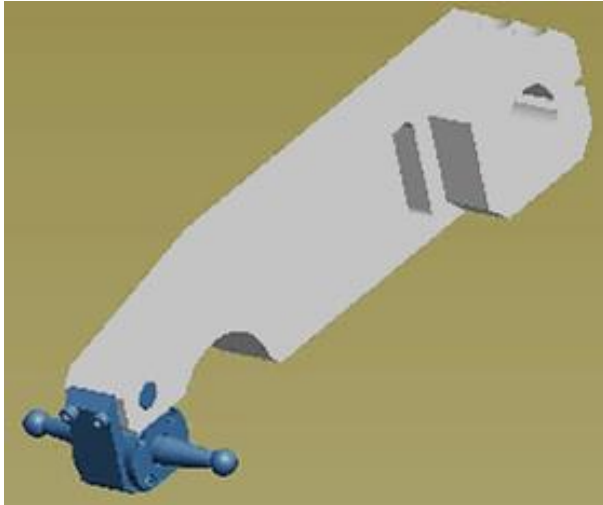
[Vidéo](#)

Source ETP-Sweden - <http://www.etp.se/>

Example of a Delta arm

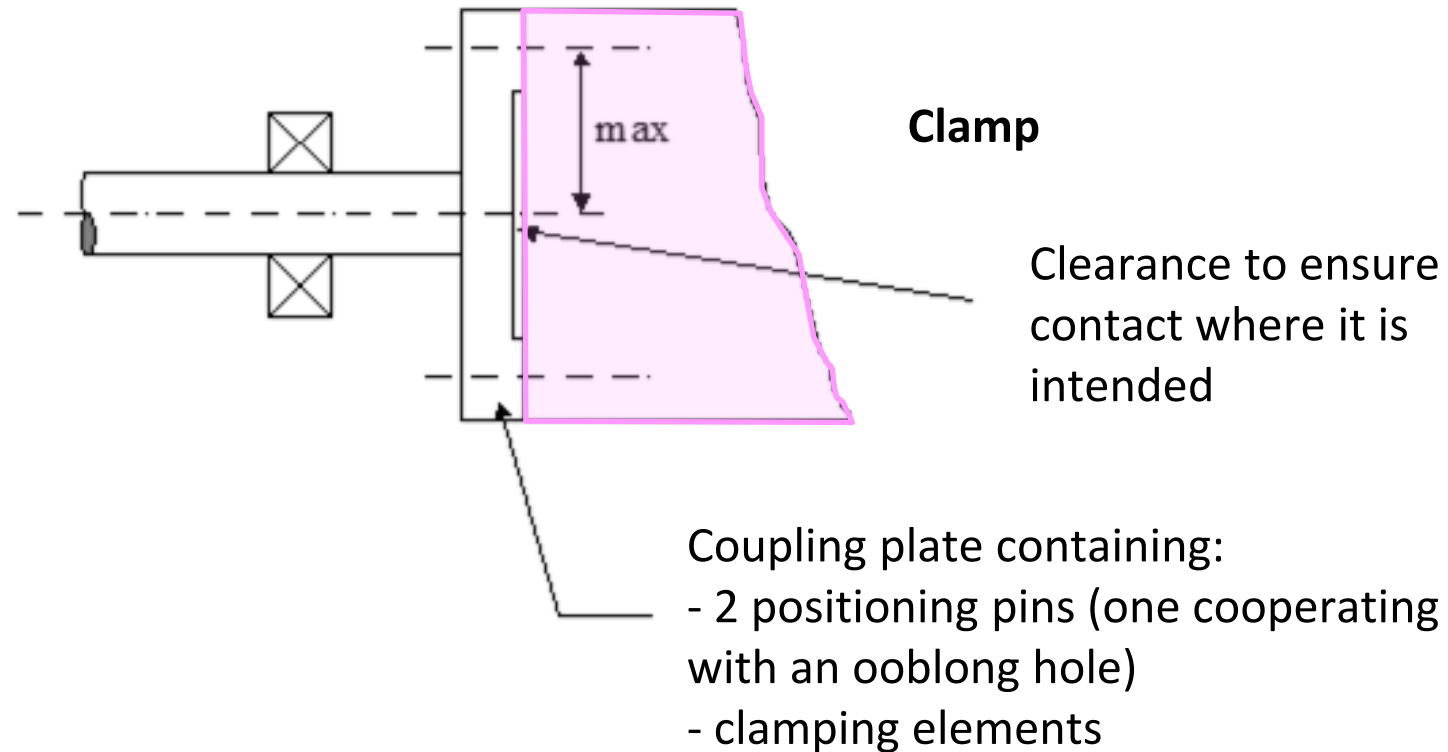


Solution by [clamping](#) : simple solution for fixing the arm on the motor shaft:



Direct coupling, screwing

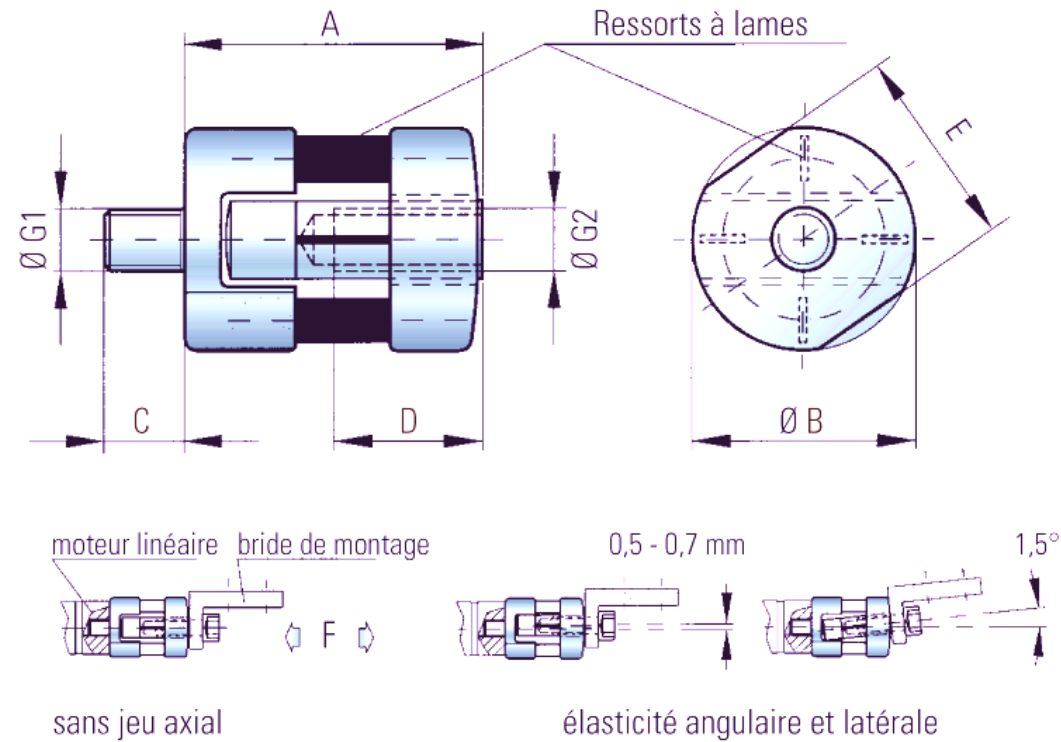
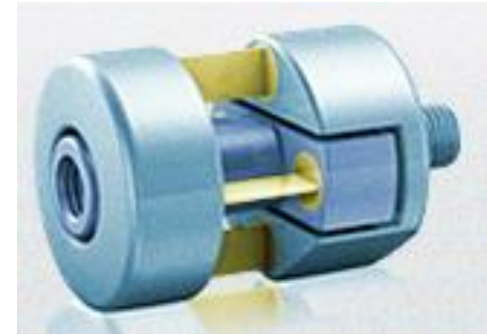
Coupling of a robot gripper by screwing



2.3 The linear bellows

Unlike the previous bellows, the linear bellows is designed to correct angular errors while being rigid in the direction of the linear movement.

Example of a design with flexible blades.



What should be considered when choosing a coupling?

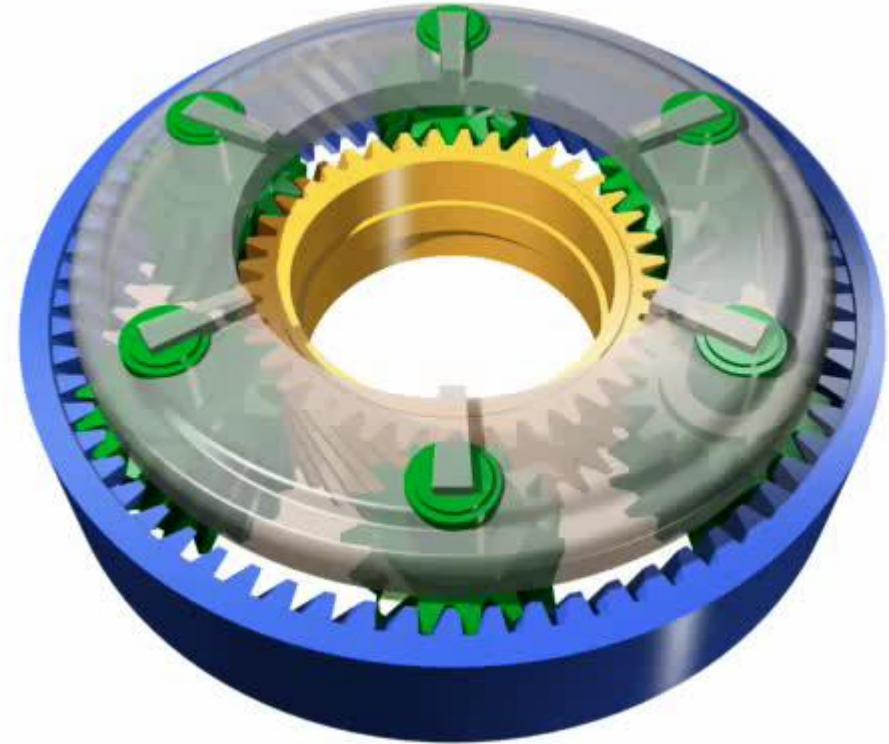
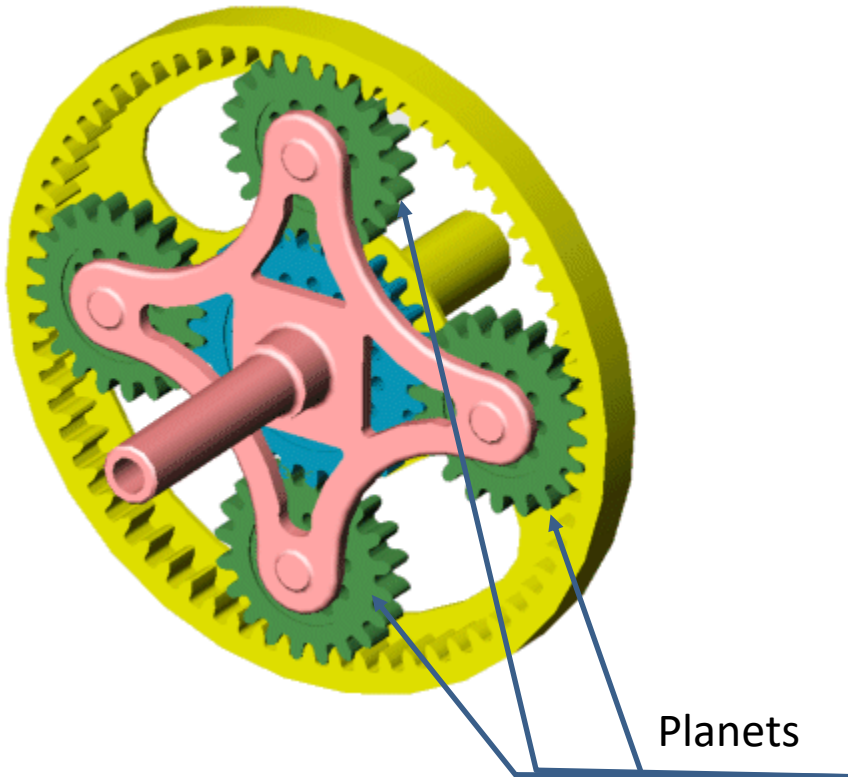
- Rigidity (we want to avoid that, the addition of this coupling weakens the system)
- Moment of inertia
- Mass
- Overall dimensions
- Transmissible torques (forces for linear movements)
- Transmissible speeds

3- Gearbox et gears

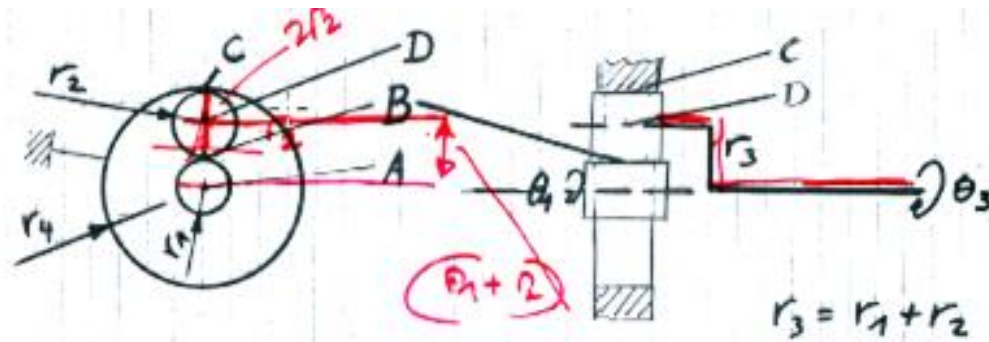
- Planetary
- Toothed belts
- Simple gears
- Left shaft and tangent screw
- Left shaft and tangent screw
- Differential gearing: Harmonic drive

3.1- Planetary gearboxes

Planetary gearboxes built with pinions (**planets**) rotating around a sun (**pinion in the center**) have several levels to achieve high gearbox ratios



3.1- The planetary gearboxes



Entrée : θ_1 Sortie : θ_3
 Rapport de transmission : $i = \frac{\theta_1}{\theta_3}$

Rapport de transmission : $i = \frac{\theta_1}{\theta_3}$

Dépl de B : $\theta_1 r_1$

$$\text{Dépl de D : } \frac{\theta_1 r_1}{2 r_2} \cdot r_2 = \frac{\theta_1 r_1}{2}$$

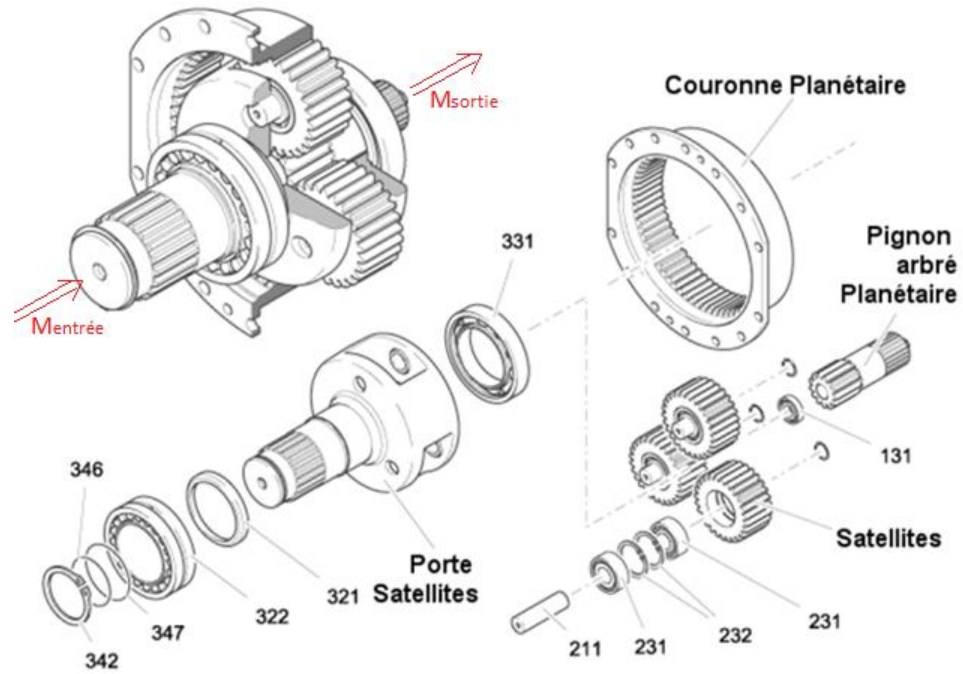
$$\theta_3 = \frac{\text{dépl de D}}{r_1 + r_2} = \theta_1 \frac{r_1}{2} \cdot \frac{1}{r_1 + r_2} = \theta_1 \frac{r_1}{r_1 + r_1 + 2r_2}$$

$\underbrace{\hspace{1.5cm}}_{r_4}$

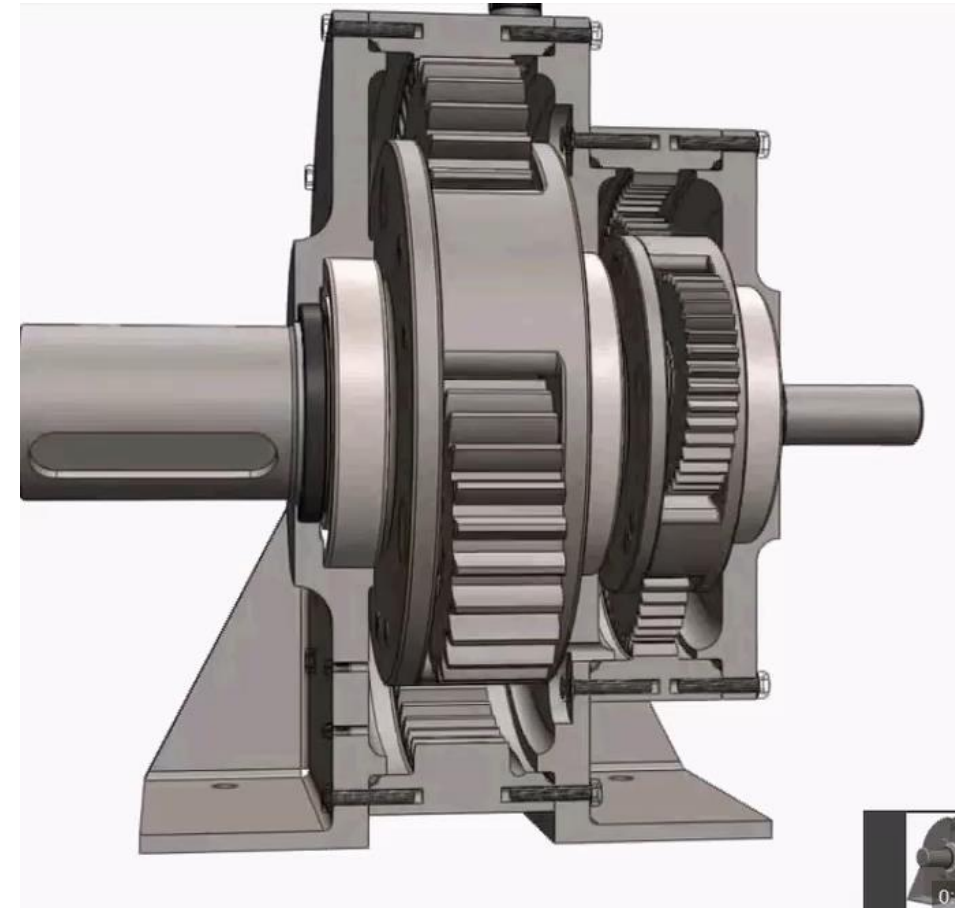
$$\theta_3 = \theta_1 \frac{r_1}{r_1 + r_4} = \theta_1 \frac{z_1}{z_1 + z_4}$$

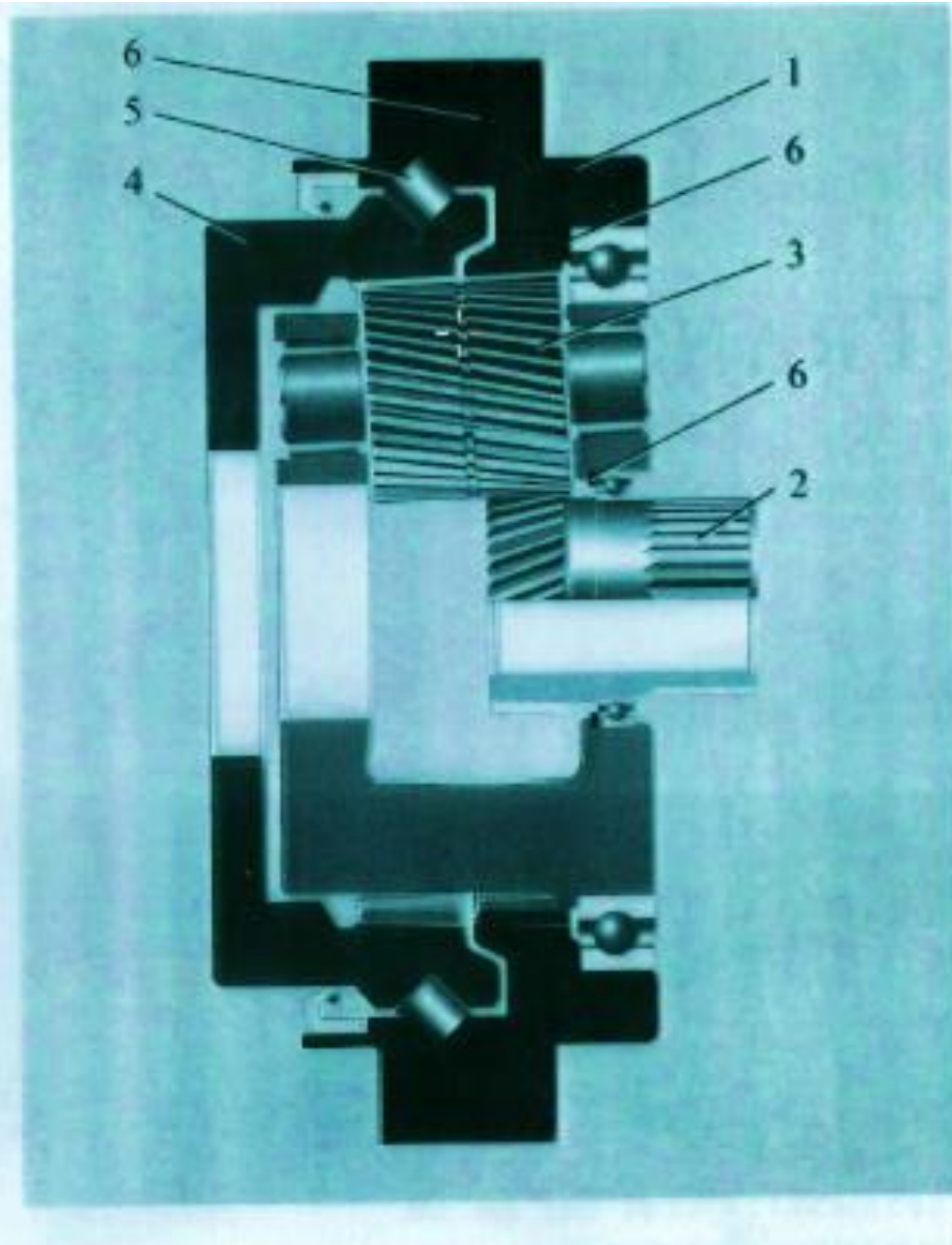
$$\boxed{i = \frac{\theta_1}{\theta_3}} = \frac{z_1 + z_4}{z_1} = \boxed{1 + \frac{z_4}{z_1}}$$

Assembly



Example of a 2-stage reductor





Transmission for main and wrist axis drives on industrial robots with handling weight between 5 and 150 kg.

The transmission includes a planetary drive of Wolfrom design with pre-ratio.

A cross-roller bearing integrated in the transmission absorbs high tilting moments and forces. Conventional external bearings are not required.

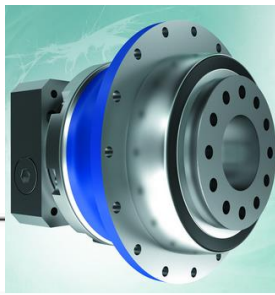
Key to drawing

- 1 Stator ring gear***
- 2 Sun gear – hollow shaft***
- 3 Step planet gear***
- 4 Output ring gear***
- 5 Cross-roller bearing***
- 6 Setting disk***

GP42 de Maxon



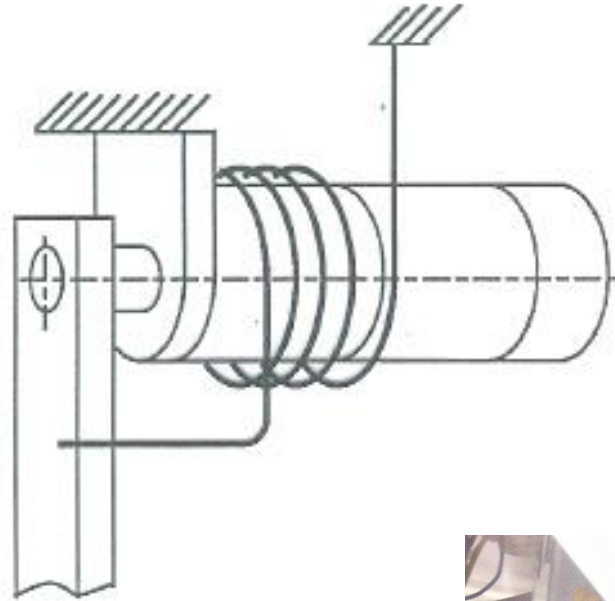
Données sur les réducteurs											
1 Rapport de réduction		3.5 : 1	12 : 1	26 : 1	43 : 1	81 : 1	156 : 1	150 : 1	285 : 1	441 : 1	756 : 1
2 Rapport de réduction exact		7/2	49/4	26	343/8	2197/27	156	2401/16	15379/54	441	756
10 Moment d'inertie	gcm ²	14	15	9.1	15	9.4	9.1	15	15	14	14
3 Diamètre max. de l'arbre du moteur	mm	10	10	8	10	8	8	10	10	10	10
Numéros de commande		203114	203116	260552*	203121	203125	260553*	203130	203134	203138	203142
1 Rapport de réduction		4.3 : 1	15 : 1	36 : 1	53 : 1	91 : 1	216 : 1	186 : 1	319 : 1	488 : 1	936 : 1
2 Rapport de réduction exact		13/3	91/6	36/1	637/12	91	216/1	4459/24	637/2	4394/9	936
10 Moment d'inertie	gcm ²	9.1	15	5.0	15	15	5.0	15	15	9.4	9.1
3 Diamètre max. de l'arbre du moteur	mm	8	10	4	10	10	4	10	10	8	8
Numéros de commande		260551*	203117		203122	203126		203131	203135	203139	260554*
1 Rapport de réduction		6 : 1	19 : 1		66 : 1	113 : 1		230 : 1	353 : 1	546 : 1	1296 : 1
2 Rapport de réduction exact		6/1	169/9		1183/18	338/3		8281/36	28561/81	546	1296/1
10 Moment d'inertie	gcm ²	4.9	9.4		15	9.4		15	9.4	14	5.0
3 Diamètre max. de l'arbre du moteur	mm	4	8		10	8		10	8	10	4
Numéros de commande			203118		203123	203127		203132	203136	203140	
1 Rapport de réduction			21 : 1		74 : 1	126 : 1		257 : 1	394 : 1	676 : 1	
2 Rapport de réduction exact			21		147/2	126		1029/4	1183/3	676	
10 Moment d'inertie	gcm ²		14		15	14		15	15	9.1	
3 Diamètre max. de l'arbre du moteur	mm		10		10	10		10	10	8	
4 Nombre d'étages		1	2	2	3	3	3	4	4	4	4
5 Couple permanent max.	Nm	3.0	7.5	7.5	15.0	15.0	15.0	15.0	15.0	15.0	15.0
6 Couple intermittent max. admissible	Nm	4.5	11.3	11.3	22.5	22.5	22.5	22.5	22.5	22.5	22.5
7 Rendement max.	%	90	81	81	72	72	72	64	64	64	64
8 Poids	g	260	360	360	460	460	460	560	560	560	560
9 Jeu moyen à vide	°	0.6	0.8	0.8	1.0	1.0	1.0	1.0	1.0	1.0	1.0
11 Longueur du réducteur L1**	mm	41.0	55.5	55.5	70.0	70.0	70.0	84.5	84.5	84.5	84.5



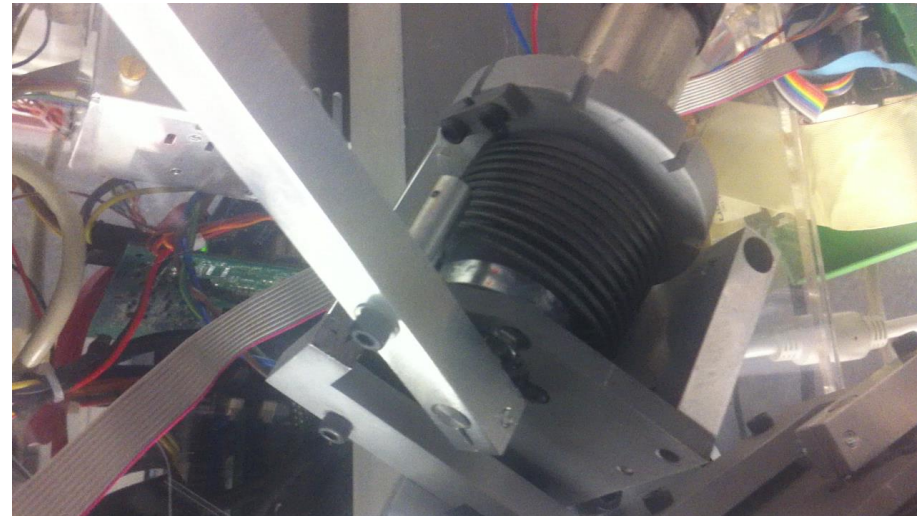
Planetary Gearboxes Wittenstein

		2-stage													
Ratio ³⁾	<i>i</i>		16	20	21	25	28	31	35	40	50	61	70	91	100
cymex [®] -optimized acceleration torque (please contact us regarding the design)	$T_{2B_{opt}}$	Nm	60	60	–	62	60	–	62	62	62	–	60	–	–
		in.lb	531	531	–	549	531	–	549	549	549	–	531	–	–
Max. acceleration torque (max. 1000 cycles per hour)	T_{2B}	Nm	55	55	40	55	55	40	55	55	55	45	55	32	35
		in.lb	487	487	354	487	487	354	487	487	487	398	487	283	310
Nominal output torque (with n_{2N})	T_{2N}	Nm	40	40	30	40	40	30	40	40	40	30	40	15	18
		in.lb	354	354	266	354	354	266	354	354	354	266	354	133	159
Emergency stop torque (permitted 1000 times during the service life of the gearhead)	T_{2Nst}	Nm	100	100	100	100	100	100	100	100	100	100	100	100	100
		in.lb	885	885	885	885	885	885	885	885	885	885	885	885	885
Nominal input speed (with T_{2N} and 20 °C ambient temperature) ²⁾	n_{2N}	rpm	4000	4000	4000	4000	4000	4000	4000	4000	4800	5500	5500	5500	5500
Max. input speed	n_{2Max}	rpm	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000
Mean no load running torque (with $n_1 = 3000$ rpm and 20 °C gearhead temperature) ⁴⁾	T_{012}	Nm	0.55	0.45	0.45	0.45	0.35	0.35	0.30	0.25	0.25	0.20	0.20	0.20	0.20
		in.lb	4.87	3.98	3.98	3.98	3.10	3.10	2.66	2.21	2.21	1.77	1.77	1.77	1.77
Max. torsional backlash	j_t	arcmin	Standard ≤ 4 / Reduced ≤ 2												
Torsional rigidity ¹⁾	C_{02}	Nm/arcmin	12	12	10	12	12	9	12	11	12	9	11	7	8
		in.lb/arcmin	106	106	89	106	106	80	106	97	106	80	97	62	71
Tilting rigidity	C_{2K}	Nm/arcmin	–												
		in.lb/arcmin													
Max. axial force ⁴⁾	F_{2AMax}	N	1630												
		lb _f	367												
Max. tilting moment	M_{2KMax}	Nm	110												
		in.lb	974												
Efficiency at full load	η	%	94												

3.1- The planetary gearboxes Compensating play



Source Murata springs



3.2- The Harmonic drive

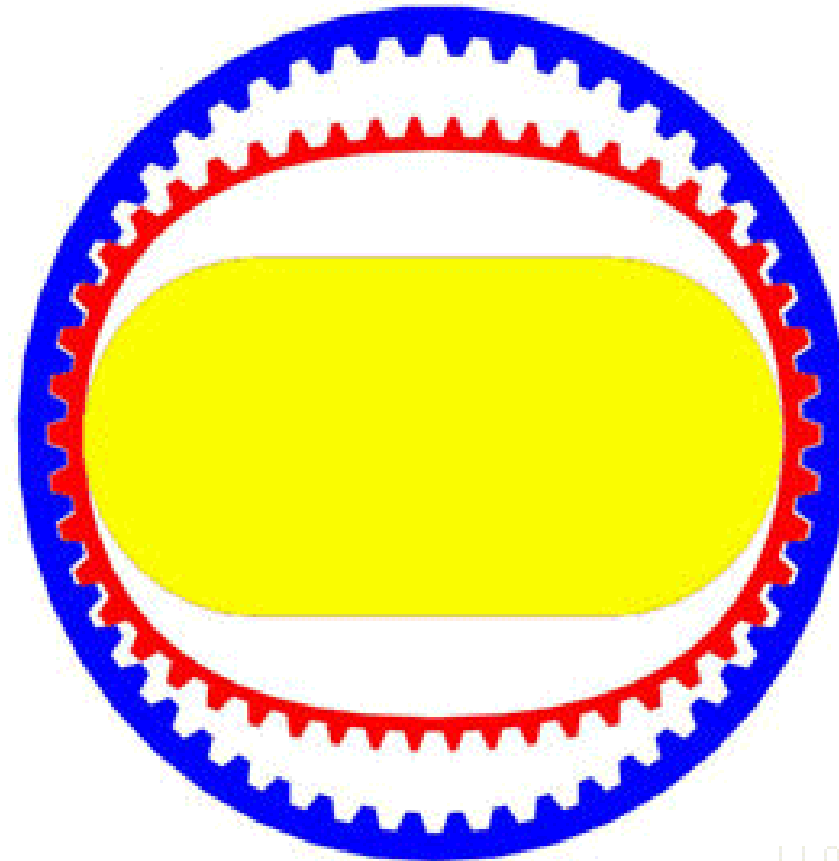
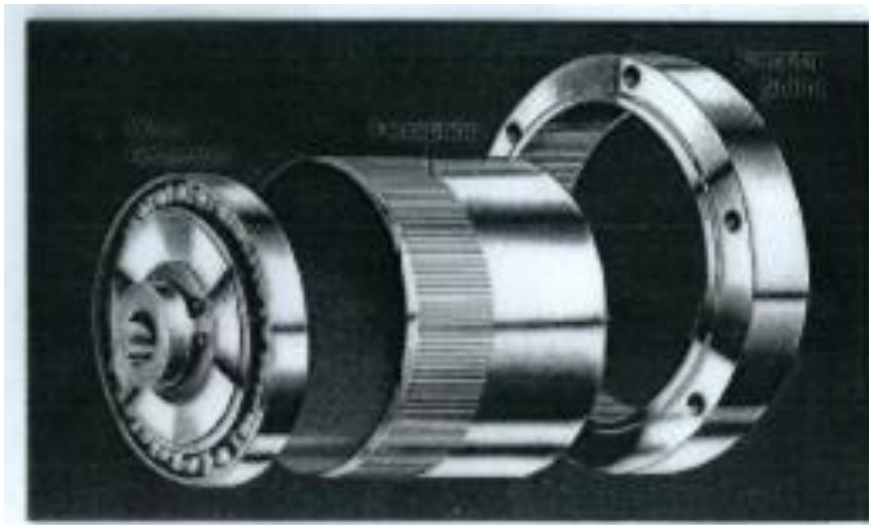
Ce type de réducteur a un rapport de transmission variant dans la fourchette ~~80~~ à 300.

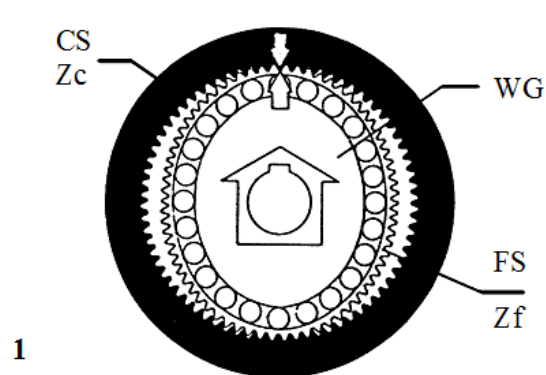
50

z_f : nombre de dents de la couronne flexible (sortie)

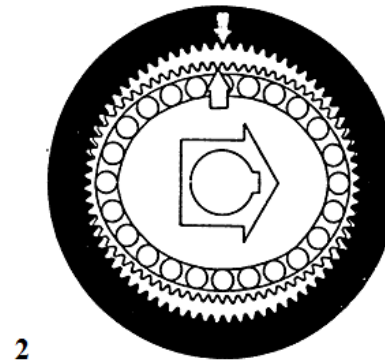
z_c : nombre de dents de la couronne rigide

$$i = \frac{z_f}{z_f - z_c}$$

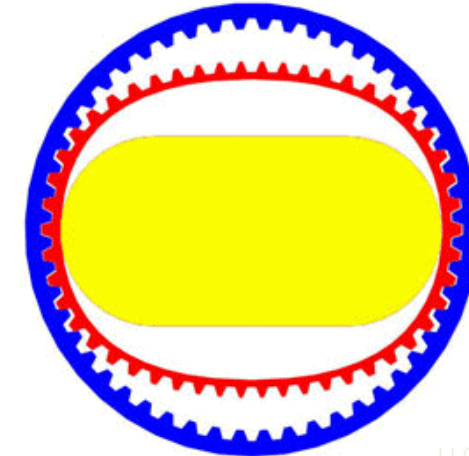




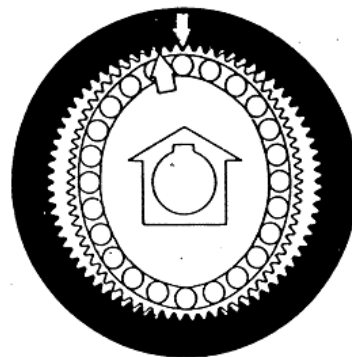
1. Le Flexspline (FS) à denture extérieure a en général 2 dents en moins que le Circular Spline (CS) et donc un diamètre primitif plus petit d'environ 1%. Il est déformé par le Wave Generator (WG) elliptique et, aux extrémités du grand axe, engrène la denture intérieure du Circular Spline



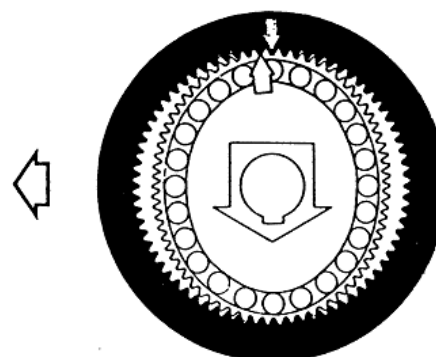
2. Lorsque le Wave Generator tourne dans le sens des aiguilles d'une montre, la zone d'engrenage se déplace avec le grand axe de l'ellipse



LL08



4. Après une rotation complète du Wave Generator, le Flexspline s'est déplacé, par rapport au Circular Spline de 2 dents dans le sens opposé.



3. Une rotation du Wave Generator de 180°, dans le sens des aiguilles d'une montre, provoque entre le Flexspline et le Circular Spline, un déplacement relatif d'une dent dans le sens opposé.

$$i = \frac{Z_f}{Z_f - Z_c}$$

Advantages – Disadvantages of the Harmonic Drive



Compact

Very high reduction by the gearbox

No play



Limited in torque and speed

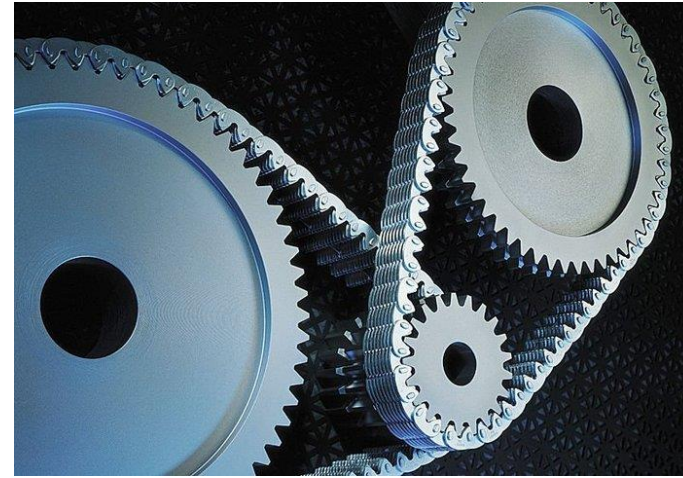
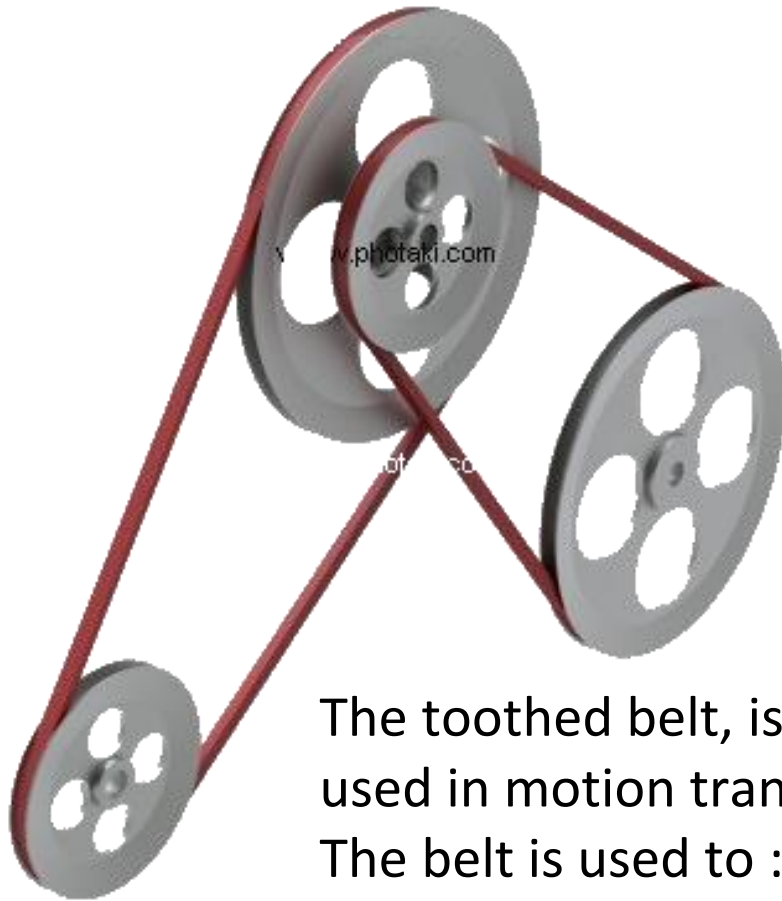
Expensive

Beware of rigidity

4- other transmissions

- The belt
- The steel band
- The cable
- The gears and the rack
- The screw nut

4.1- Transmissions with toothed belts



The toothed belt, is one of the cheapest components used in motion transmission.

The belt is used to :

- Transmitting rotation (**rotation-rotation**)
- Transmitting translation (**rotation-translation**)

4.1- Transmissions with toothed belt

Rotation-Rotation

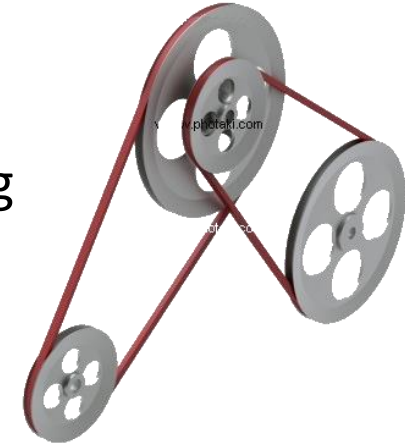
The belt drive allows a simple and inexpensive coupling between two parallel shafts.



- The alignment of the shafts (input and output) is no longer a problem.
- The use of several stages allows for a greater gain.

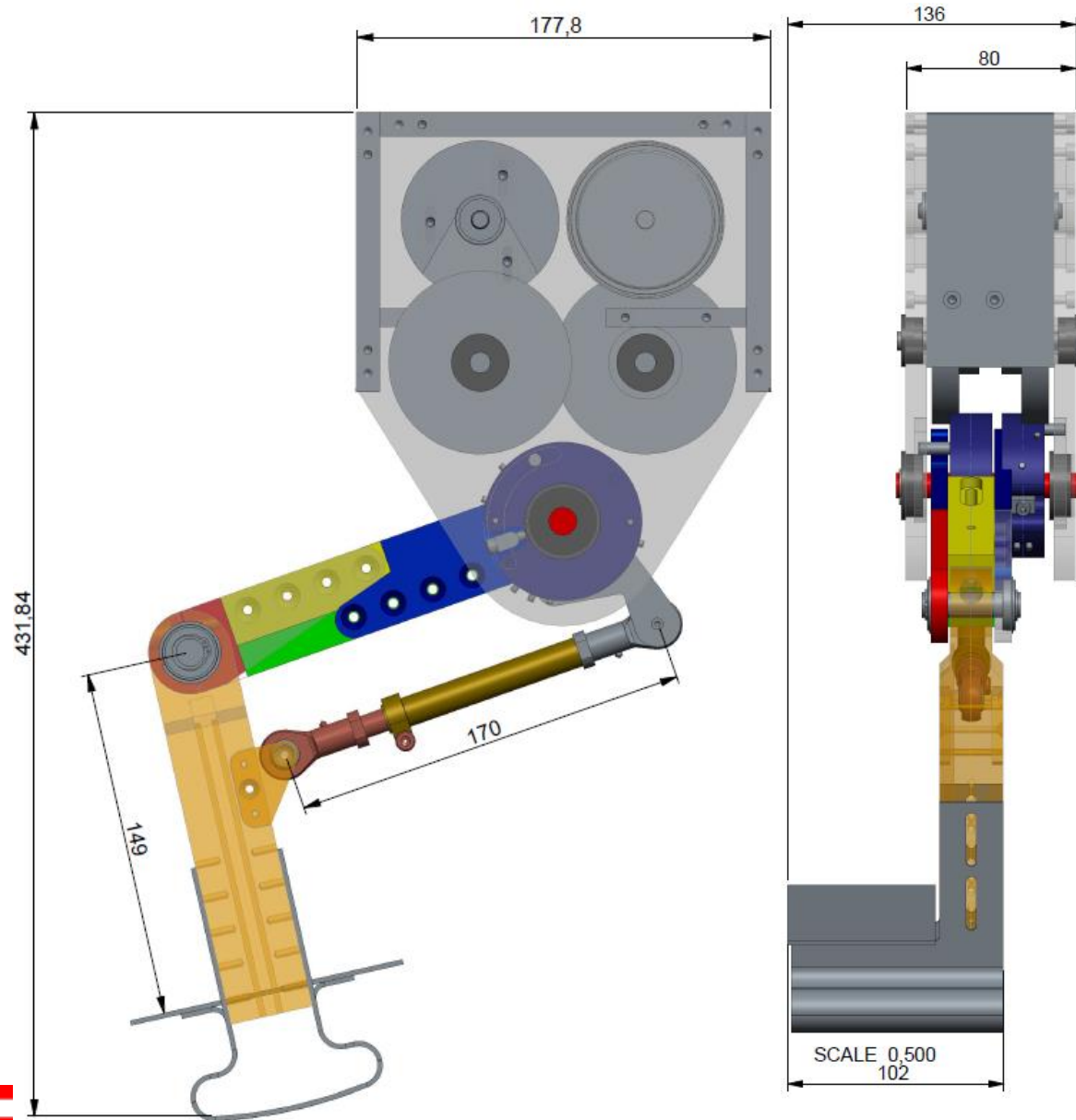


- Attention must be paid to the belt preload to avoid backlash.
- The rigidity of the belt is disadvantageous to the rigidity of the transmission if it is long.
- The gearbox ratio depends on the diameter of the pulleys.



4.1- Transmissions with toothed belts

Example Rotation-Rotation



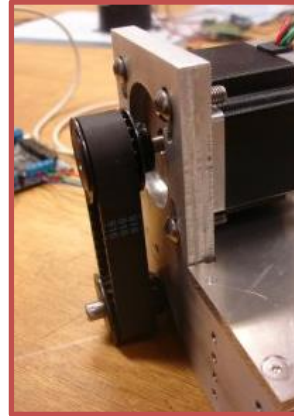
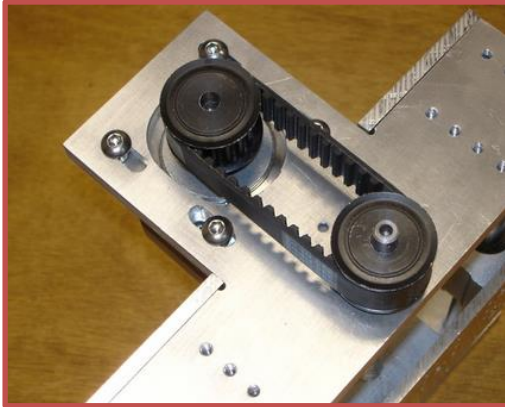
2 stages of reduction.
Using motors with 150mNm.
Total gear ratio = $5 \times 7 = 35$
Available output torque = 15Nm



4.1- Toothed belt drives

Example 2 & 3 - Rotation-Rotation

The belt is also used as a **coupling element** between a drive screw and a motor shaft



The advantage of this arrangement is that it prevents the need for a bellows-type coupling element while rearranging and saving space



4.1- Toothed belt drives

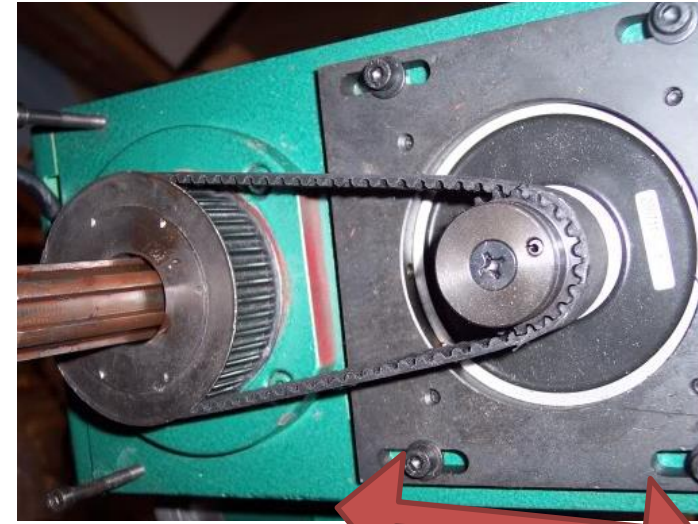
Belt preloading

Simple method:
Motor on mounting plate
and **oblong holes**

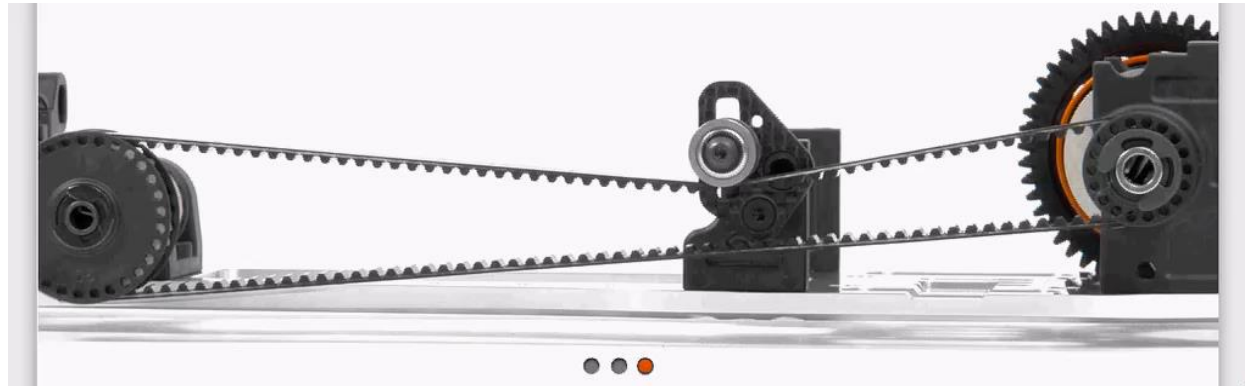


Pull the motor to preload the belt

By using a tensioner (e.g. roller) if the belt is longer.

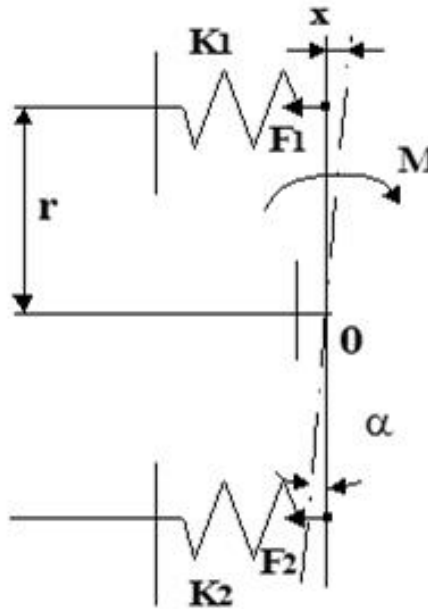


Pull the motor to preload the belt



4.1- Toothed belt drives

Calculation of the angular stiffness of a belt drive



Pulley of a belt transmission.

The 2 strands are considered as 2 pre-stressed springs (pre-tension of the belt)

$$M=0 \rightarrow F_1=F_2=F \neq 0$$

$$\alpha=0$$

Angular stiffness considering that the belt is blocked on the 2nd pinion and neglecting the effect of the wrapped part of the belt.

$$K_{\alpha} = \frac{dM}{d\alpha}$$

$$F_1 = F + K_1 \cdot dx$$

$$F_2 = F - K_2 \cdot dx$$

$$dx = d\alpha \cdot r$$

$$dM = (F_1 - F_2) \cdot r = d\alpha \cdot r^2 \cdot (K_1 + K_2)$$

$$K_{\alpha} = r^2 \cdot (K_1 + K_2)$$



$$K_{\alpha} = 2 \cdot K \cdot r^2$$

4.1- Toothed belt drives

Rotation-Translation

This solution is « The cheapest way to achieve linear motion ».

Advantages:



High speeds possible.
Low cost of transmission.

Disadvantages:



It is almost a direct drive solution. The transmission ratio Rotation - Translation is only linked to the diameter of the pulley. This implies a low position resolution (! Precision) and a low force available at the output.



Pay attention to the belt stiffness: it induces a position error due to exerted forces!

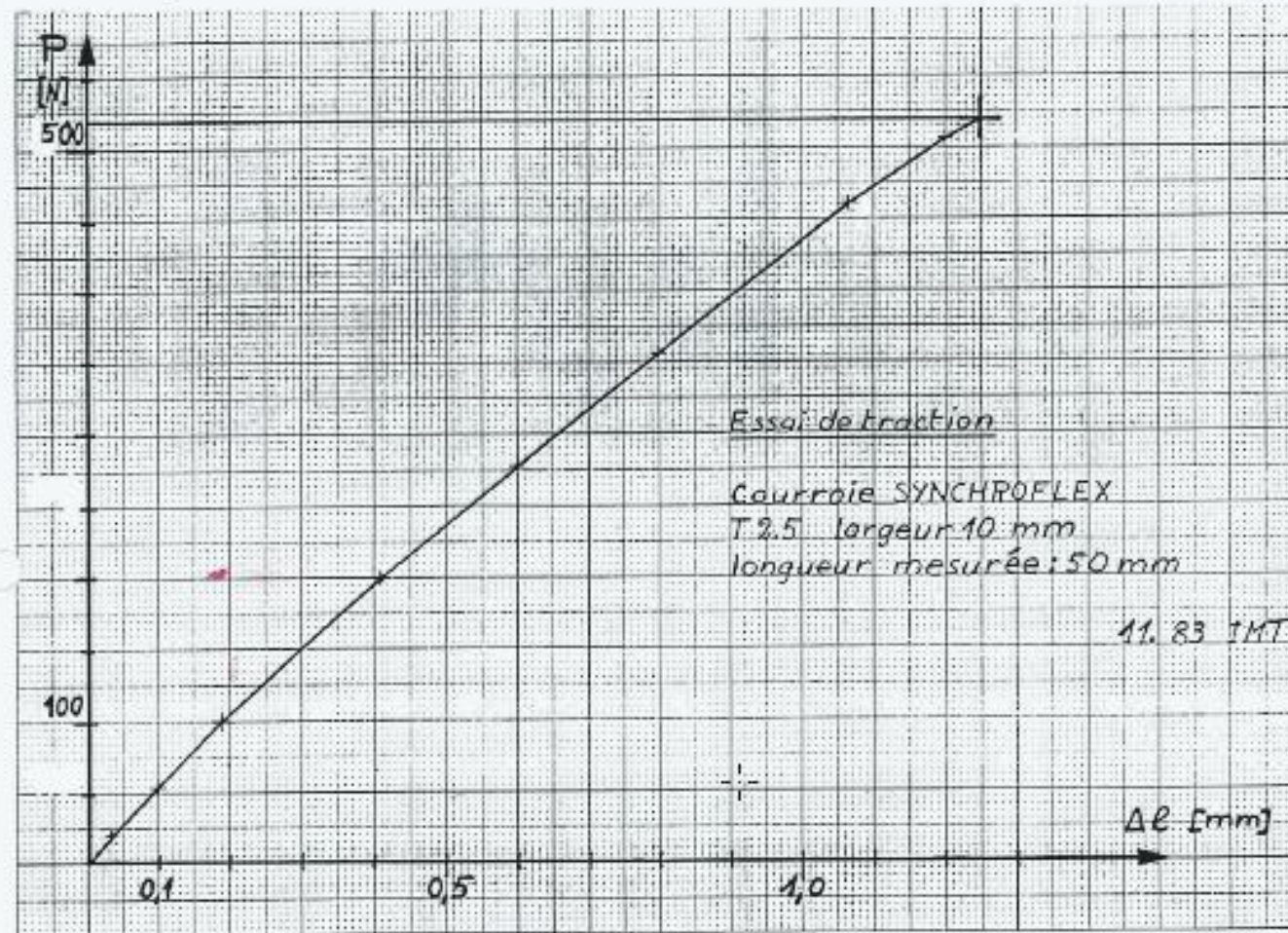


Fig. 24 Caractéristique élastique d'une courroie dentée SYNCHROFLEX T 2,5, largeur 10 mm.
Rupture pour $P = 520$ N
Limite d'utilisation donnée par le fabricant : 100 N
 $K = 24 \cdot 10^3 / \ell$ (N/m) ℓ : longueur du brin



The Linear Delta - Horizontal variant

- Belt transmission

Exercise:

Consider a linear axis of **500 mm stroke** made by a direct belt drive and using:

- Two 10mm diameter pulleys (one on each side of the axis).
- A motor equipped with an encoder of 1000 holes/turn.
 - Maximum speed of the motor is 6500 rpm.
 - Nominal torque is 120 mNm.

Questions:

1. What is the max linear speed?
2. What is the linear resolution?
3. What would be the nominal force available at the carriage?
4. Comment on your results compared to an 8mm wide T5 belt from Mulco!

Toothed belt drives (Synthesis):

Advantages:



light weight, **no stroke limitation**, easy backlash compensation, easy assembly, adaptation to the required transmission stiffness by varying the belt width and the \varnothing of the pulleys, durable, large selection available;

Disadvantages:



Rather average stiffness, average precision due to **gearing**, possible **hard** points at tooth entry (especially with high preloading);



Basic rule:

Work with pulleys of higher diameters, taking care to lighten them as much as possible (material, hollow); this rule is also valid for any mechanical transmission.

4.2- Steel band transmissions



Transmission by flexible element **docked on the drums**; several turns on each drum allow to avoid limiting the movement to 180°. A: Docking point.

Advantages:

light weight, stiffness, easy backlash compensation, **no stroke** limitation if used as a belt (sliding possible or perforated belt);

Disadvantages:

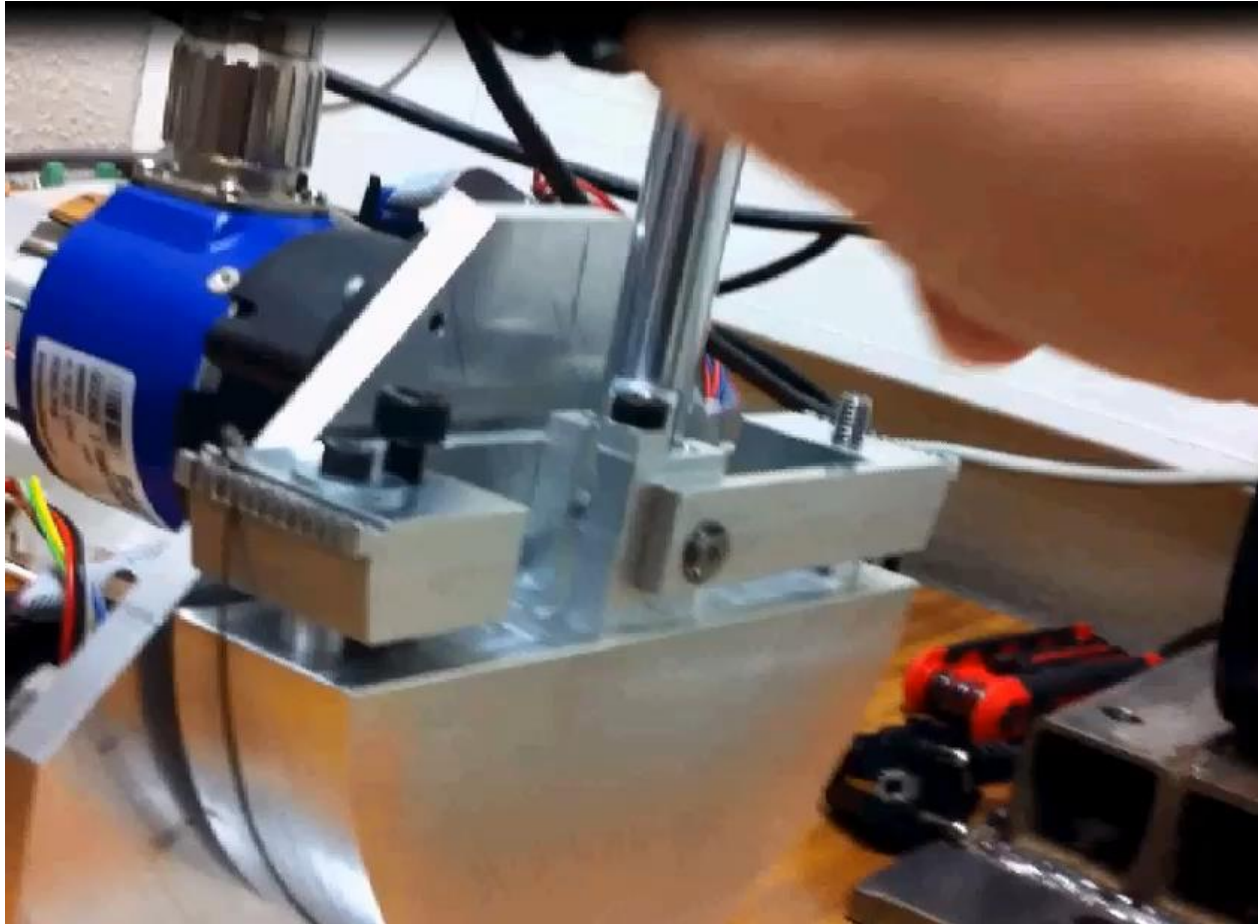
manufacturing accuracy problems and gearing accuracy problems if perforated band; variable length if wrapped on a drum > 360°; high bending stress:

$$\sigma = \frac{E \cdot e}{D}$$

if thickness e is small \Rightarrow delicate band; a band shaped like a sabre blade; this causes high stress and lack of stiffness.

This explains why **steel band transmissions are rarely used in robotic drives.**

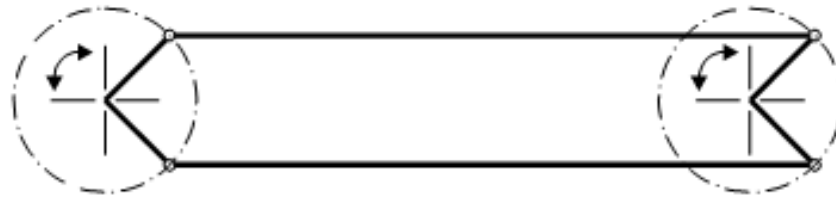
4.4- Cable transmission



- More rigidity than a belt
- Must also be **docked**
- Very good efficiency
- Slippage avoided by capstan effect

4.5- Connecting rod transmissions

Des bielles et leviers liés par des articulations sur corps roulants permettent de transmettre des mouvements sur des distances assez importantes (1 à 2 m) sans jeu, avec une faible inertie et une raideur élevée. Cette transmission ne permet pas des réductions ou des amplifications de couples. Ce genre de construction impose souvent une limitation de course.



Δ - bielles de longueur identique
- \varnothing bielles \equiv entr'axe

Fig. 21 Transmission de mouvement par bielles et manivelles. (Si le mouvement est limité à 120° , une seule bielle bien placée peut suffire).

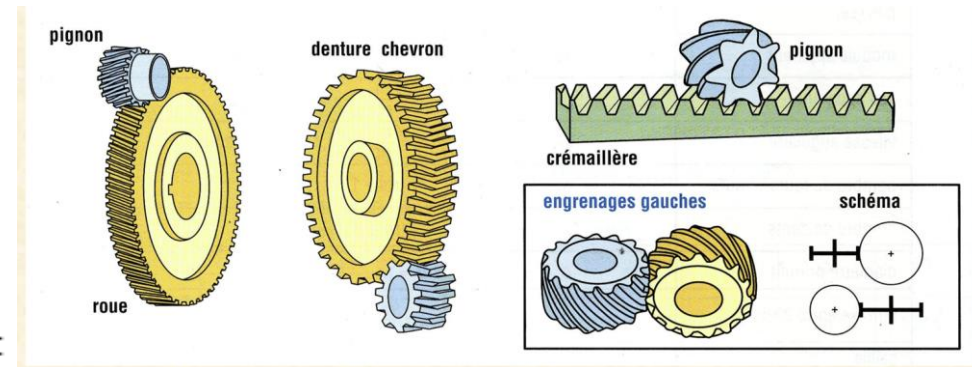
Le mouvement n'est pas limité si les manivelles sont disposées à la manière d'un vilebrequin.

La raideur d'une telle transmission est variable fonction de la position.

Il est nécessaire de contrôler la résistance des bielles au flambage, vu qu'elles sont chargées aussi bien en compression qu'en traction.

4.5- Gears - Screws - Racks

Les principales formes rencontrées sont les suivantes:



Ref: zpag.net

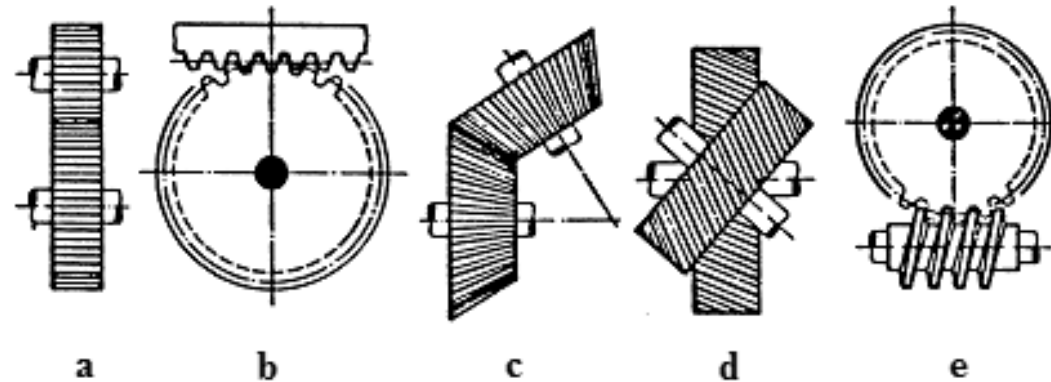


Fig. 27

Différentes formes d'engrenages:

- a) engrenage cylindrique à dents droites (dents hélicoïdales ou à chevrons)
- b) pignon-crémaillère (cas particulier de a)
- c) engrenage conique
- d) engrenage cylindrique à denture hélicoïdale et axes gauches
- e) vis tangente et couronne (cas particulier de d)

Par le fait qu'aucune pièce n'est parfaite géométriquement, l'engrenage doit fonctionner avec un certain jeu; celui-ci est néfaste par les imprécisions et les chocs qu'il provoque; il faut donc le réduire ou l'éliminer (voir Fig. 29).

Axes parallèles: (pignon-crémaillère compris)

- **dents droites:**

Avantages: - fabrication facile

Inconvénients: - interférence pour $z < 17$
- bruit

- **dents hélicoïdales :**

Avantages: - le pignon peut comporter moins de dents (≈ 10)
avant l'apparition d'interférence

Inconvénients: - moins de bruit (meilleur recouvrement)
- génère une force axiale / plus difficile à fabriquer.



Les engrenages à axes // permettent la réalisation de réducteurs planétaires;



Axes concourants:

Engrenages coniques obligatoires;

Inconvénients:

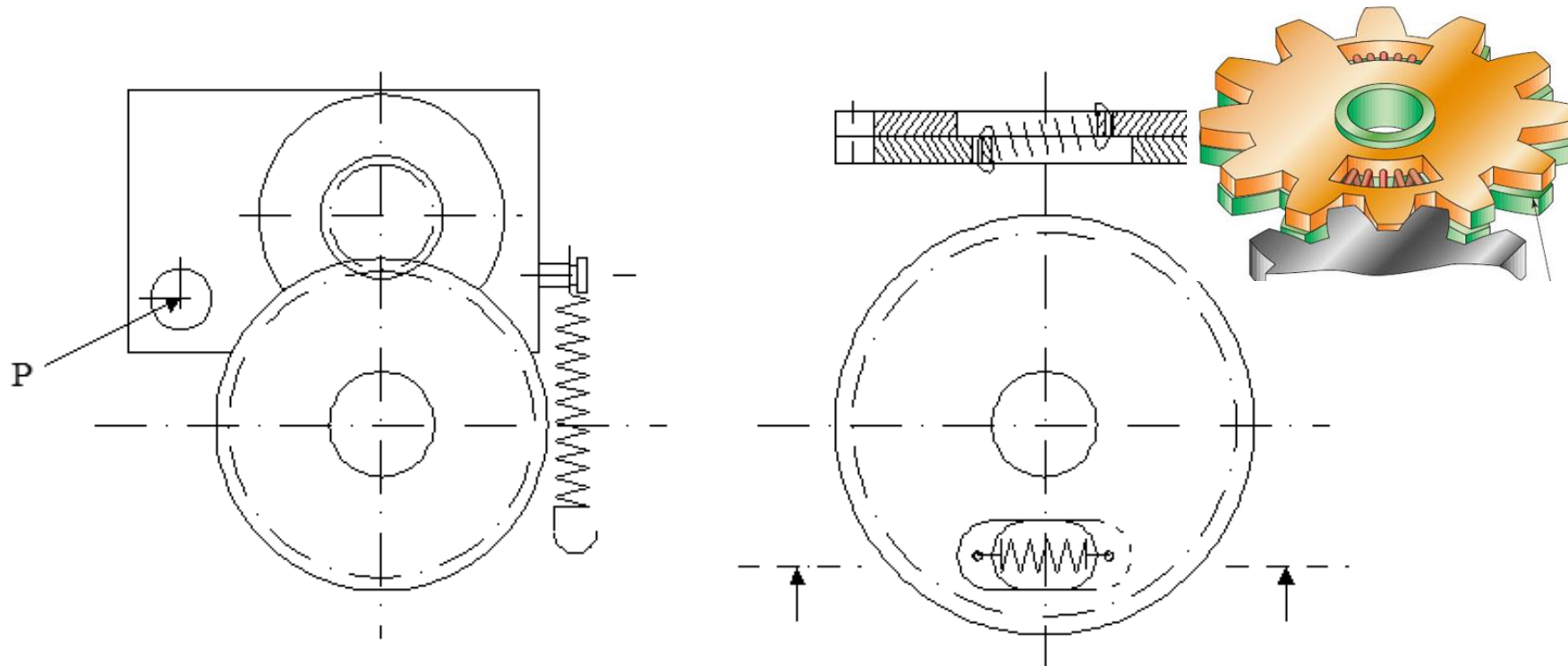
- difficiles à usiner
- jeu difficile à rattraper à cause du module variable tout au long de la dent.

Cette forme d'engrenage est principalement rencontrée au niveau de poignets de RI

Adjusting the backlash in a gear

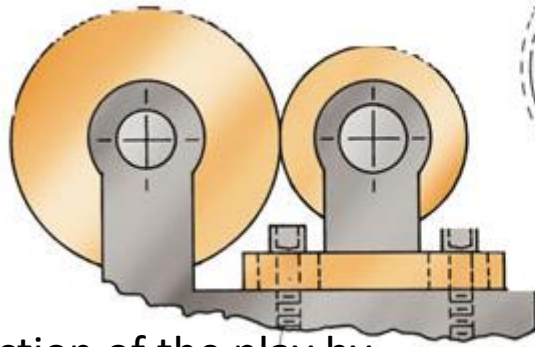
Le rattrapage peut être effectué en agissant sur l'entr'axe (dents poussées radialement les unes contre les autres) ou en coupant en 2 parties une roue de l'engrenage par un plan perpendiculaire à l'axe et en donnant un couple tendant à décaler angulairement les dents de ces 2 demi-roues.

Ce dernier moyen est également applicable aux engrenages coniques.

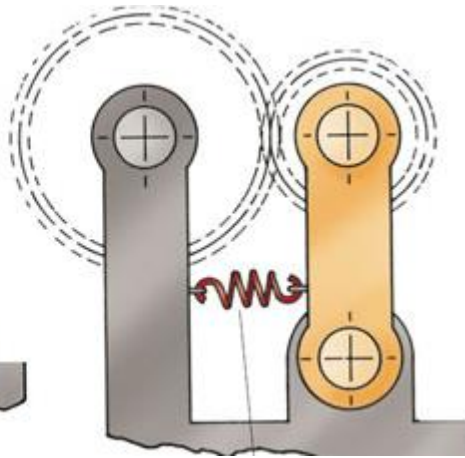


These 2 principles are also valid for the rack and pinion assembly.

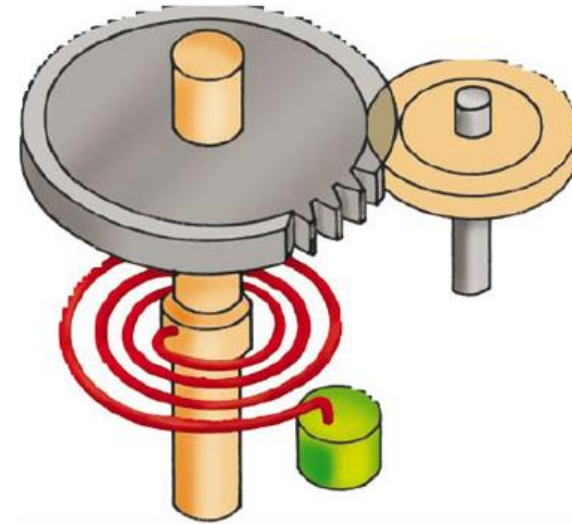
Rattrapage de jeu dans un engrenage



Reduction of the play by adjusting the distance between the wheels (e.g. by slotted hole)



Backlash reduction by means of preloading spring



Use of a spiral spring

Exercise –

Consider gear A,B with backlash adjustment (spring K_3)-

Gear B is the gear to be meshed-

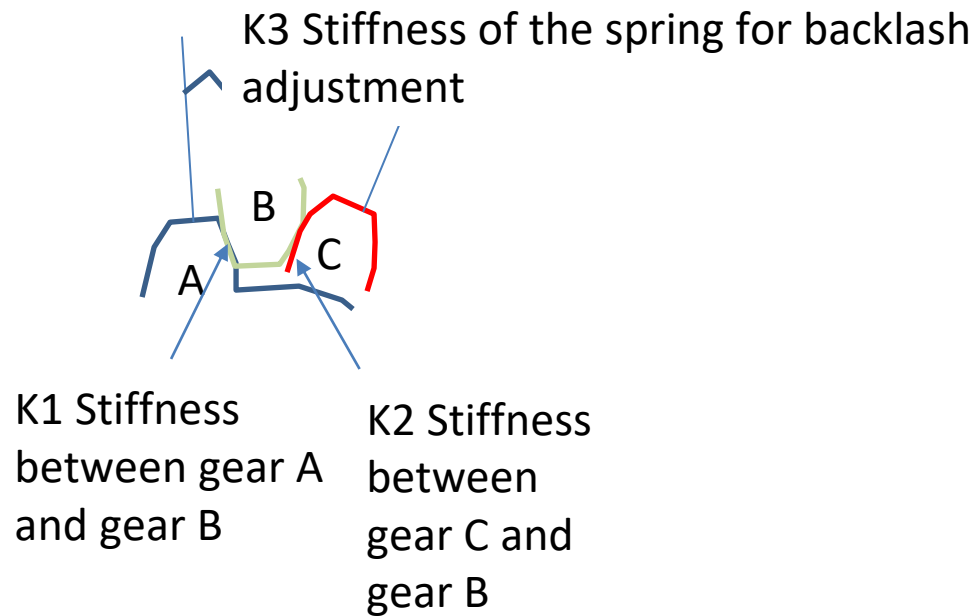
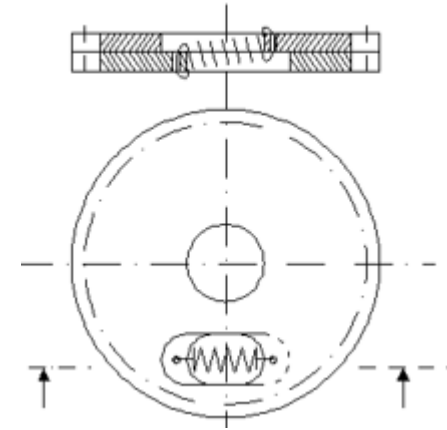
K_1 is the stiffness between gear A and gear B

K_2 is the stiffness between gear C and gear B

Considering wheel A as reference-

>> Make the equivalent spring diagram.

What is the equivalent stiffness of the system?



The screw-nut transmission

For the implementation of linear motion.

- Can achieve very high output forces (widely used in machine tools)
- Reasonable cost.
- The backlash is controlled to have very high precision.
- Good rigidity for lengths $< 1.0\text{m}$

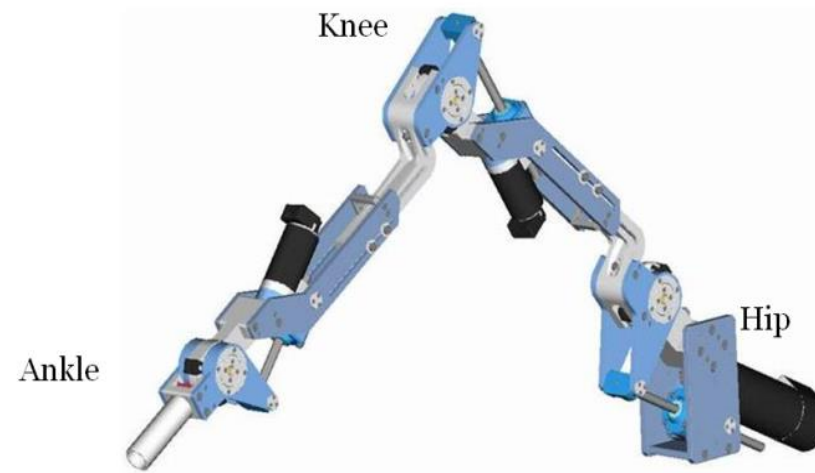


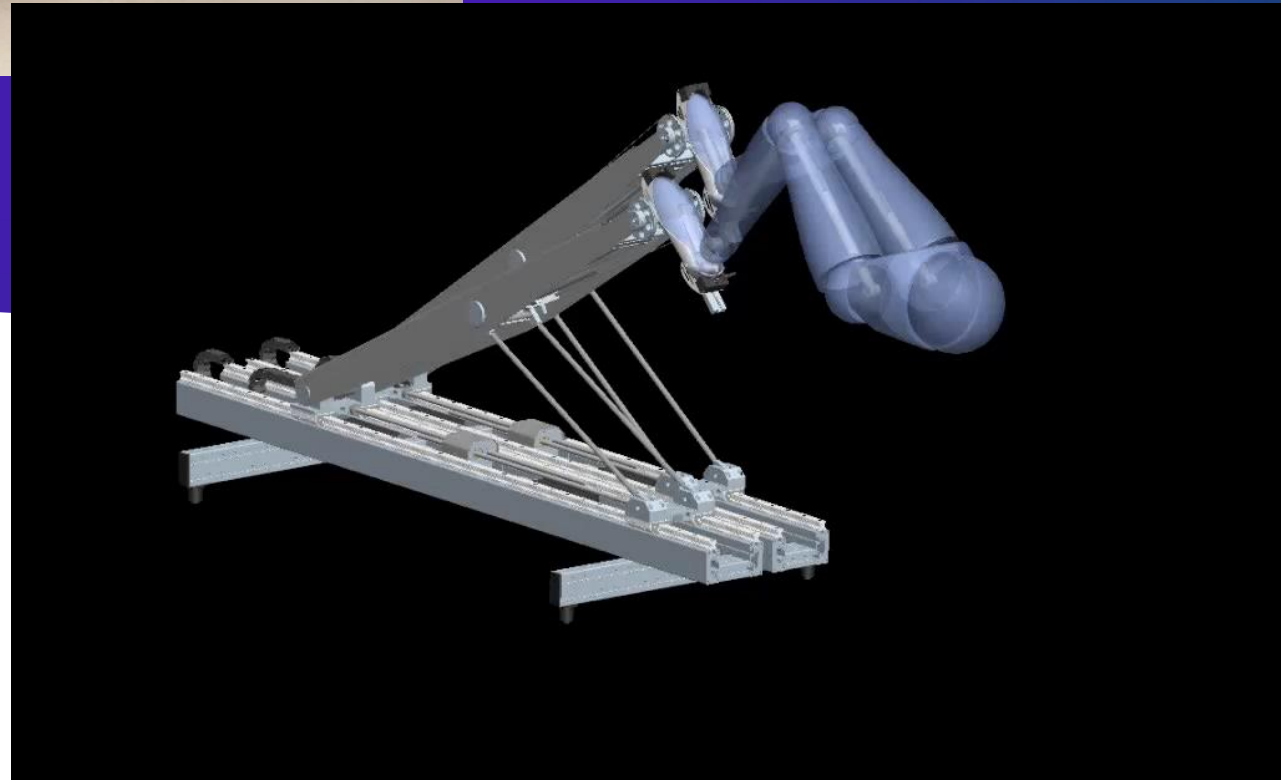


The Minagle –
Ref. Flexible Hinges realization

The Rehabilitation device MotionMaker

- Screw driven angular joint





The screw-nut transmission

Implementation

avec ou sans éléments roulants

sans: rendement idem engrenages
hélicoïdaux (filet carré);
si filet trapézoïdal ou triangulaire:
 $\mu' = \frac{\mu}{\cos \alpha}$ $\alpha = \frac{1}{2} \times$ sommet.

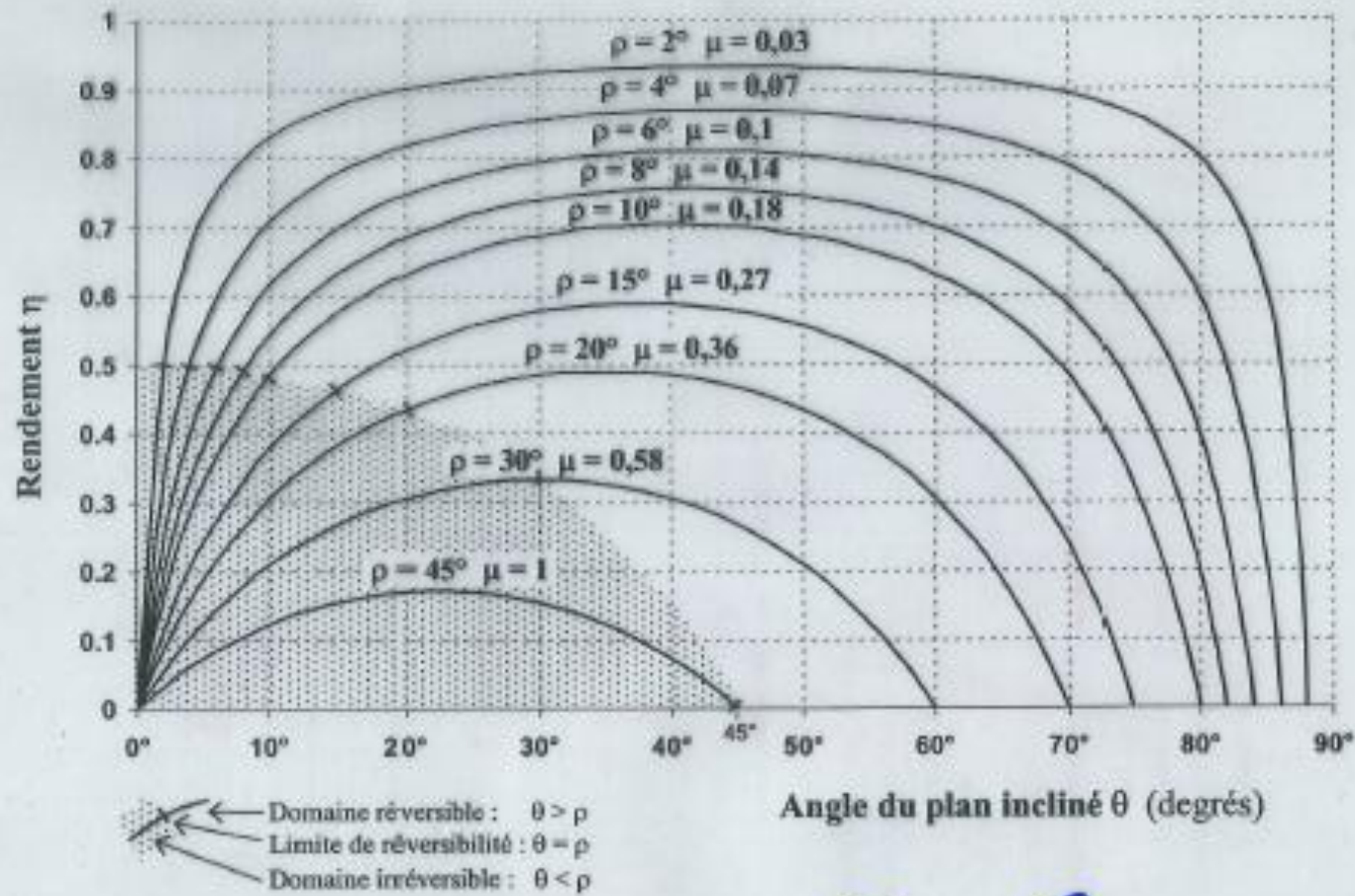
Toujours du jeu si pas
d'éléments élastiques ou
contre-écrou.

avec: circulation de billes ou rouleaux
satellites.

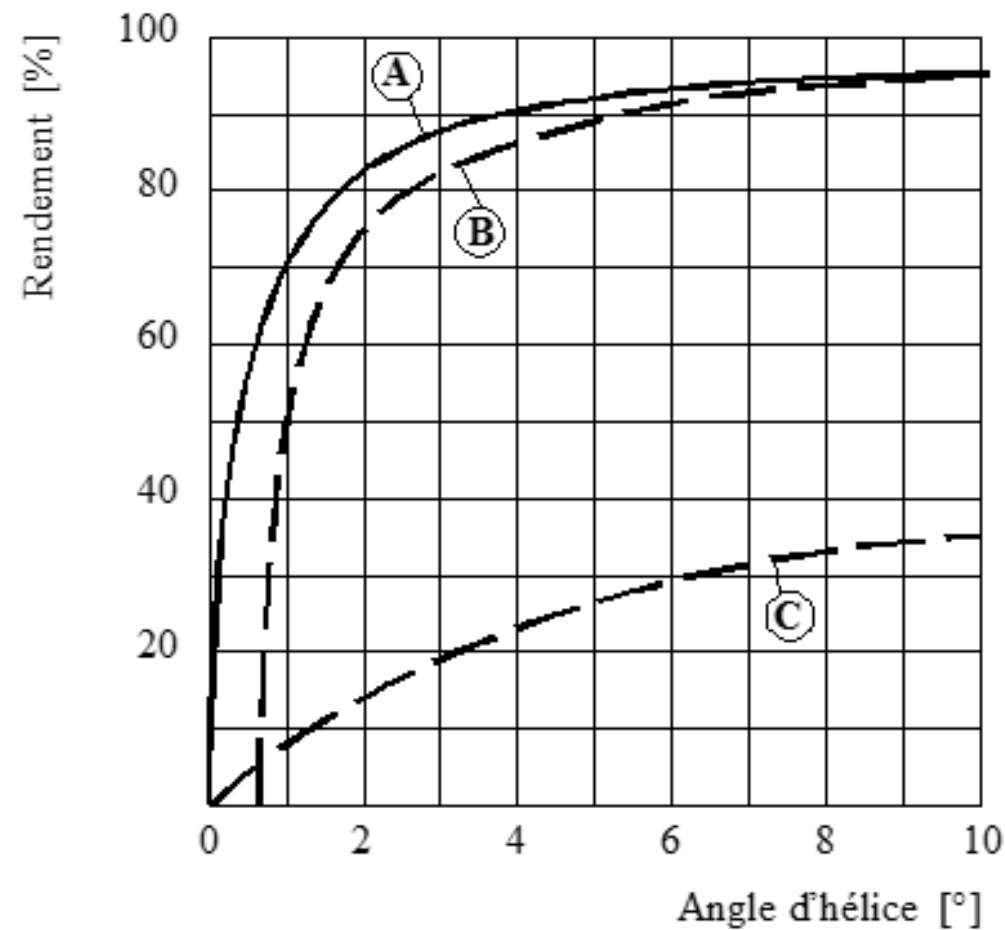
Jeu axial vis à billes: 0,1 à 0,2mm



Fig. 3.26 Rendement du plan incliné $\eta = \frac{\tan \theta}{\tan (\theta + \rho)}$ pour différentes valeurs de ρ



$$\eta_{\max} \text{ pour } \theta = \frac{\pi}{4} - \frac{\rho}{2}$$



Rendement de transmissions à vis (Données Warner).

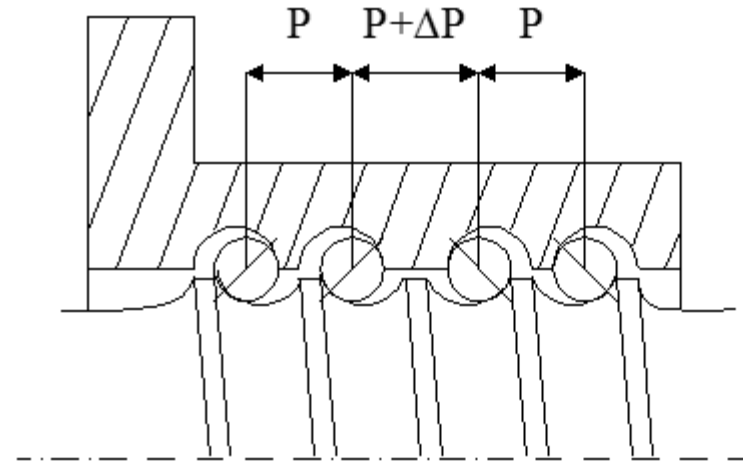
- A) Vis à billes avec l'énergie apportée par la vis.
- B) Vis à billes avec l'énergie apportée par l'écrou.
- C) Vis à filetage trapézoïdal

The screw-nut transmission

Backlash adjustment

Le jeu d'une vis à billes est éliminé par l'utilisation de 2 écrous précontraints ou par l'utilisation d'écrous spéciaux dont le pas est augmenté entre les 2 spires centrales.

Fig.33 Simple écrou à billes précontraint (Warner)



La vitesse de rotation d'une vis est limitée en fonction de sa longueur et de son diamètre (vitesse critique limitée par la fréquence propre de la vis).

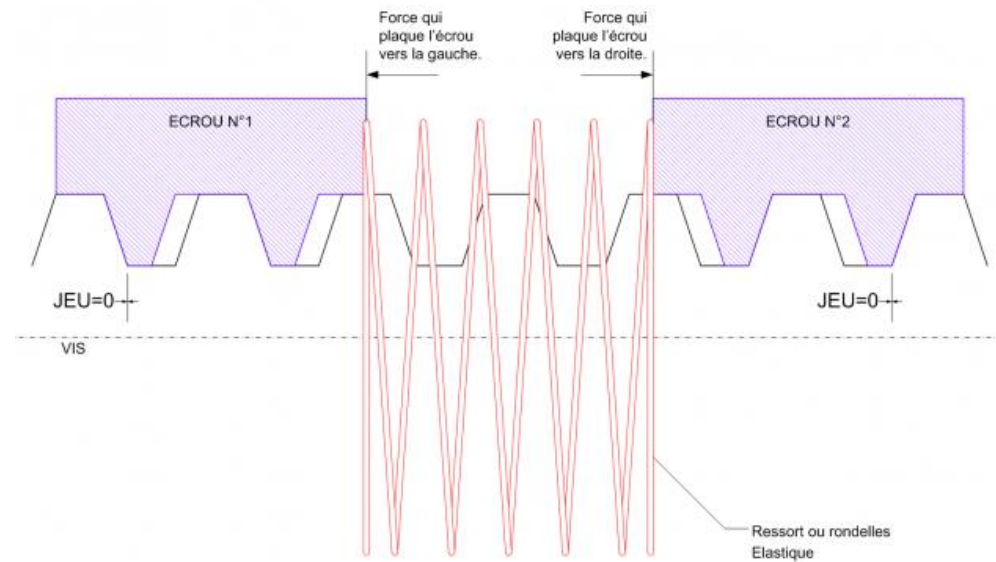
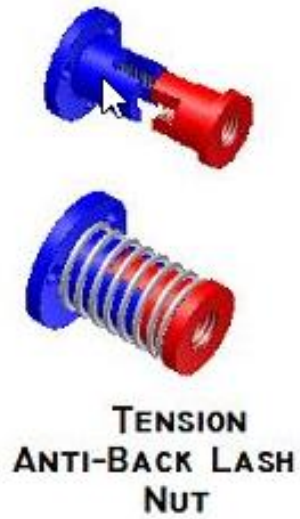
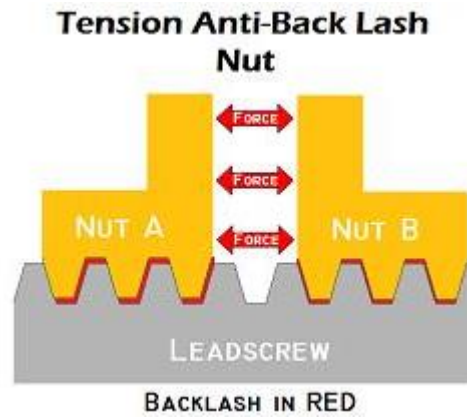
Ainsi une vitesse de déplacement d'un mobile élevée demandera un pas de vis important, donc une réduction faible.

Exemple



The screw-nut transmission

Backlash adjustment



Carl Bruno Strandgren Moved to Epalinges in 1956 Died in 1984



Inventaire - Vis 2 nouveaux s'ajoutent - 1942
↳ Volonté - 1962

Carl Bruno Strandgren est né le 6 avril 1894 à Umea, petite ville côtière au nord de la Mer Baltique; il s'établit à Goeteborg à l'âge de 15 ans où il fréquente l'Ecole d'ingénieurs. Mobilisé en 1915 par la Suède, il est affecté comme stagiaire artilleur dans différentes usines où il se familiarise avec les machines de l'époque. En 1920 il s'établit à Paris et mène la vie modeste d'étudiant en suivant les cours de mathématique et de philosophie du Collège de France.

